AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 42 No. 556. **AUGUST, 1952** PRICE: 3s. 6d.

Perfection

The 'Star of Egypt'

... and "Chromidium" for Cylinders

Par Excellence

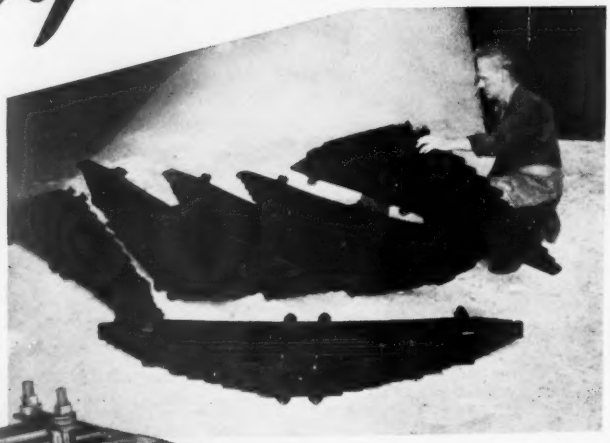
MIDCYL

"Chromidium" is the exclusive product of:—

THE MIDLAND MOTOR CYLINDER CO. LTD., BIRMID WORKS, SMETHWICK
BIRMINGHAM 40

M-W56

Mighty Springs for

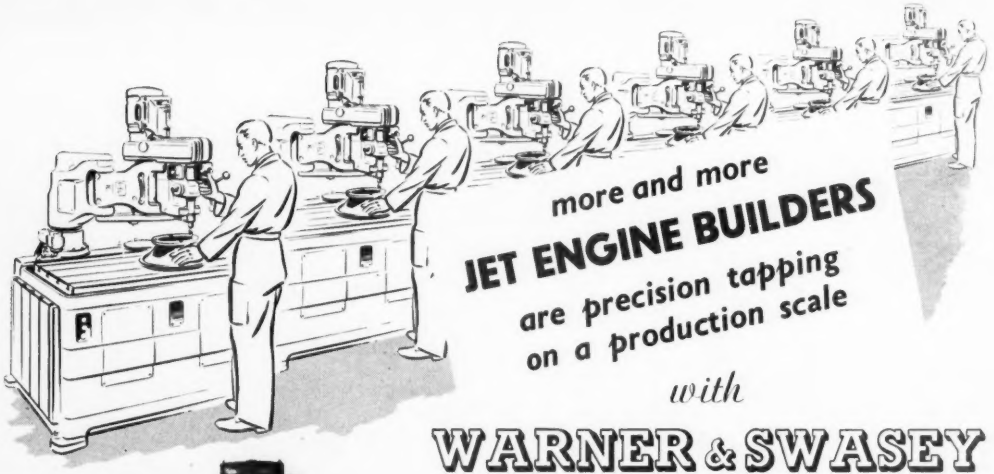


The Mighty "THORNYCROFT" ANTAR

Believed to be the heaviest ever made in this country, these Springs are 62" long by 5" wide and weigh almost 7 cwts. each and are included in the four wheel driving bogies supplied by the Kirkstall Forge Engineering Limited to Transport Equipment (Thornycroft) Limited, the builders of this powerful vehicle.

ENGLISH STEEL CORPORATION LTD

SHEFFIELD



more and more

JET ENGINE BUILDERS

are precision tapping on a production scale

with

WARNER & SWASEY

RADIAL TAPPING & THREADING MACHINES

Here's the machine for threading large bulky work on a production scale with speed and precision. Constant uniformity of threads is assured by a new leadscrew and guide finger method of leading on the tap or die. Tap life is prolonged, backlash and excessive pressure are eliminated and taps or dies are positively controlled on both threading and retracting strokes.

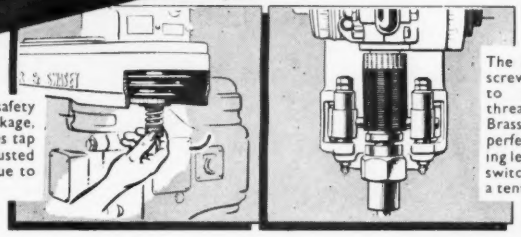
The No. 11R illustrated has three separate working surfaces that can be used simultaneously, and it can be supplied as a duplex machine with two independently-operated heads if required.

Tapping Capacities

Pipe threads in aluminium, soft bronze, brass and similar materials	¼" to 1"
Pipe threads in cast iron, mild steel, nickel and chrome-moly. steels	¼" to 1"
Threads in aluminium, plastics or soft material ..	3 B.A. to ⅝"
Threads in mild steels	3 B.A. to ⅝"
Threads in nickel and chrome-moly. steels	3 B.A. to ½"

Working Surfaces

Top, 28" × 46½" Left end vertical 33" × 24½" Rear vertical, 35½" × 31½"



An extremely sensitive safety clutch prevents tap breakage, reduces rejects and increases tap life and it can be quickly adjusted to deliver the correct torque to the tap.

The upper portion of the leadscrew is a hob which is employed to reproduce the precision threads of the leadscrew on the Brass Guide Fingers ensuring a perfectly mated fit and eliminating lead backlash and drag. A limit switch controls depth to within a tenth of a revolution of the tap.

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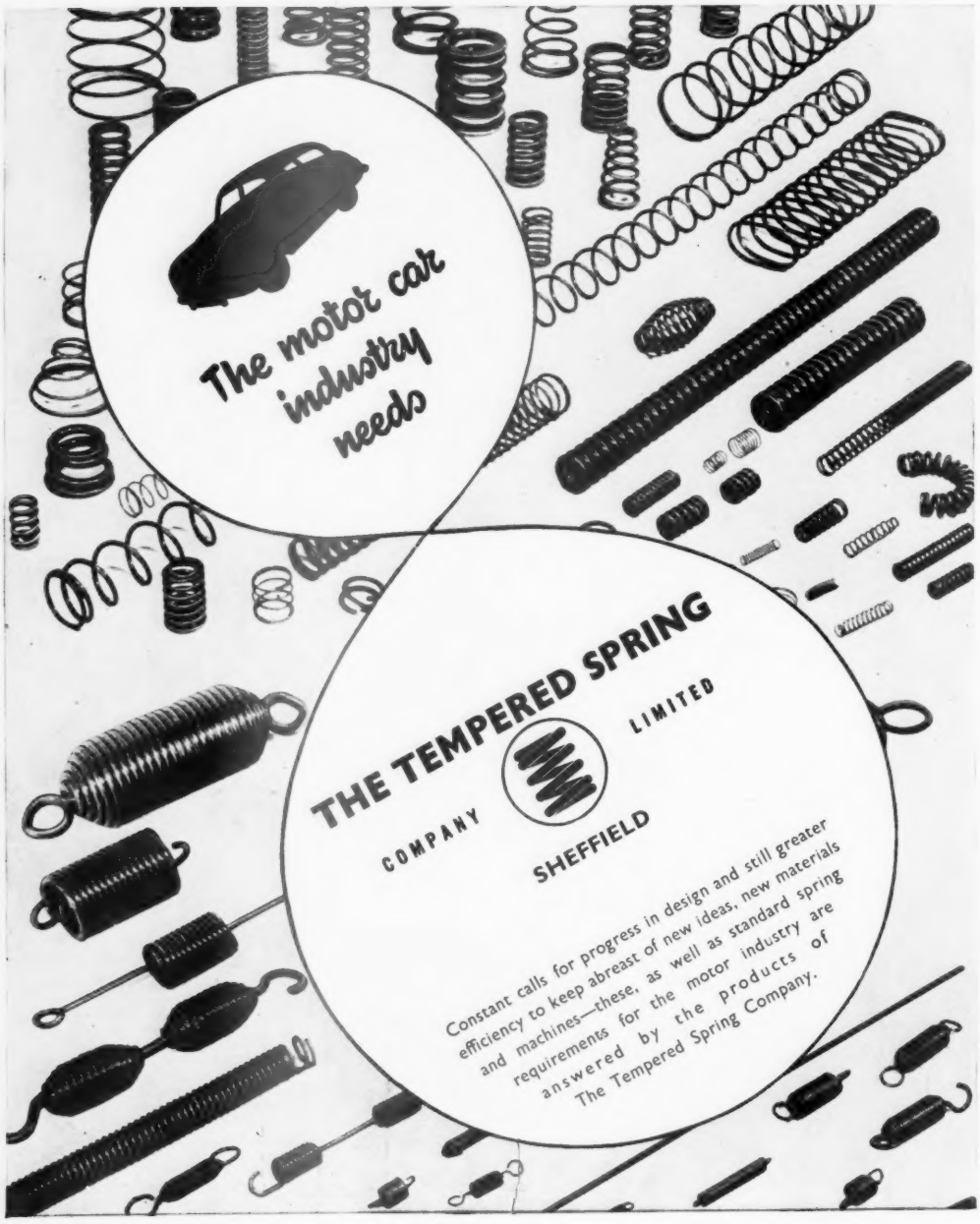
The motor car industry needs
THE TEMPERED SPRING COMPANY LIMITED
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Constant calls for progress in design and still greater efficiency to keep abreast of new ideas, new materials and machines—these, as well as standard spring requirements for the motor industry are answered by the products of The Tempered Spring Company.

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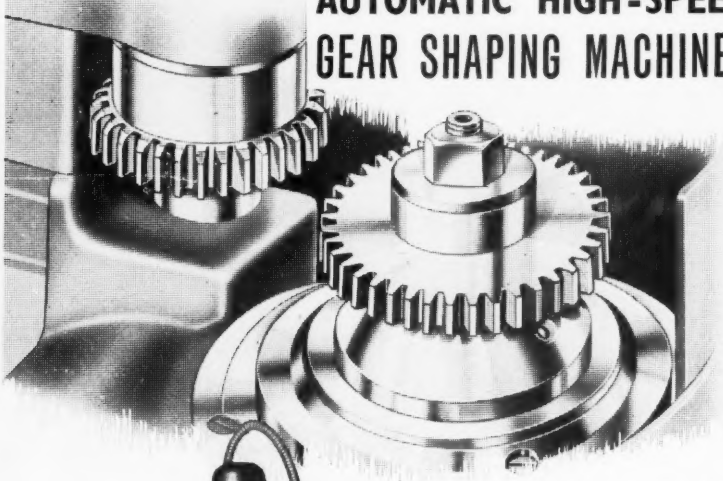
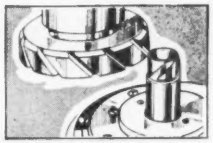
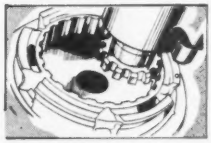
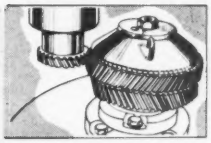
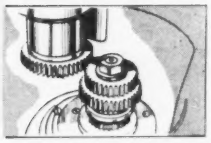
AUTOMATIC HIGH-SPEED GEAR SHAPING MACHINES

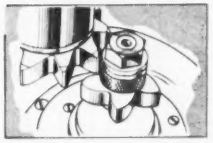


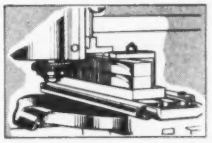


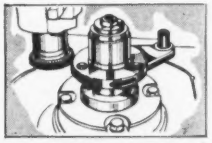


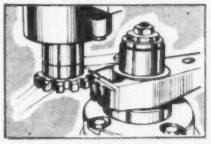


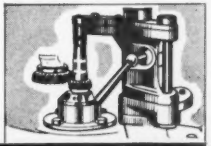








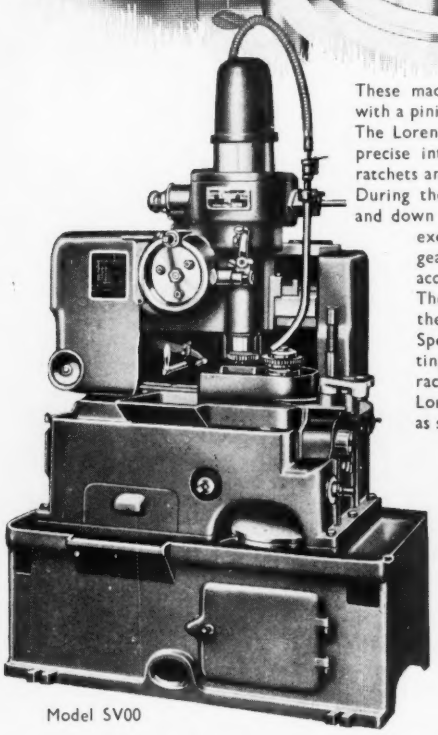




These machines shape by the generating process with a pinion type cutter for spur and helical gears. The Lorenz method is ideal for the production of precise internal and external gearing, sprockets, ratchets and other profiles.

During the cutting stroke the tool is moved up and down and rotated so that the tool and work execute the movement of two meshing gears. This method ensures maximum accuracy and smooth flanks of the work. The tool is re-sharpened by simply grinding the tooth face.

Special attachments are available for generating helical gears, gear trains, face gears, rack gears, shaping between centres, etc. Lorenz gear shapers are made in five sizes as shown below.



Model SV00

AUTOMATIC GEAR SHAPERS

MODEL	S 00	SV 00	S 5	S7/500	S7/1000
Pitch diam.	7½"	7½"	17½"	19½"	40"
Face width	2"	2"	3"	5½"	6.5/16"
Diam. pitch/module	6/4	6/4	5/3	3.6/7	3.6/7

AUTOMATIC GEAR HOBBERS

High production machines of extreme accuracy for the manufacture of spur, helical, worm wheels, and special gearings are made for work of pitch diameters from 23½" to 63" and diam. pitch/module 4.23/6 to 1.59/16.

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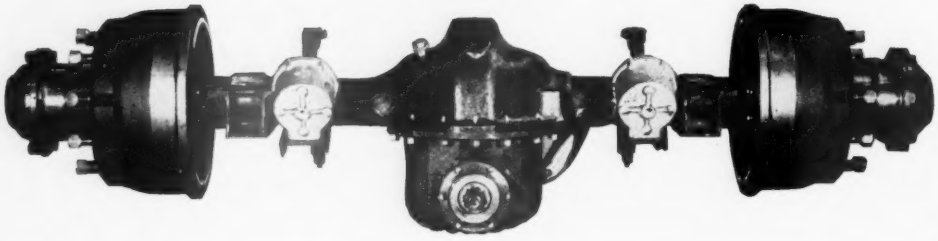

KIRKSTALL FORGE ENGINEERING LTD
LEEDS

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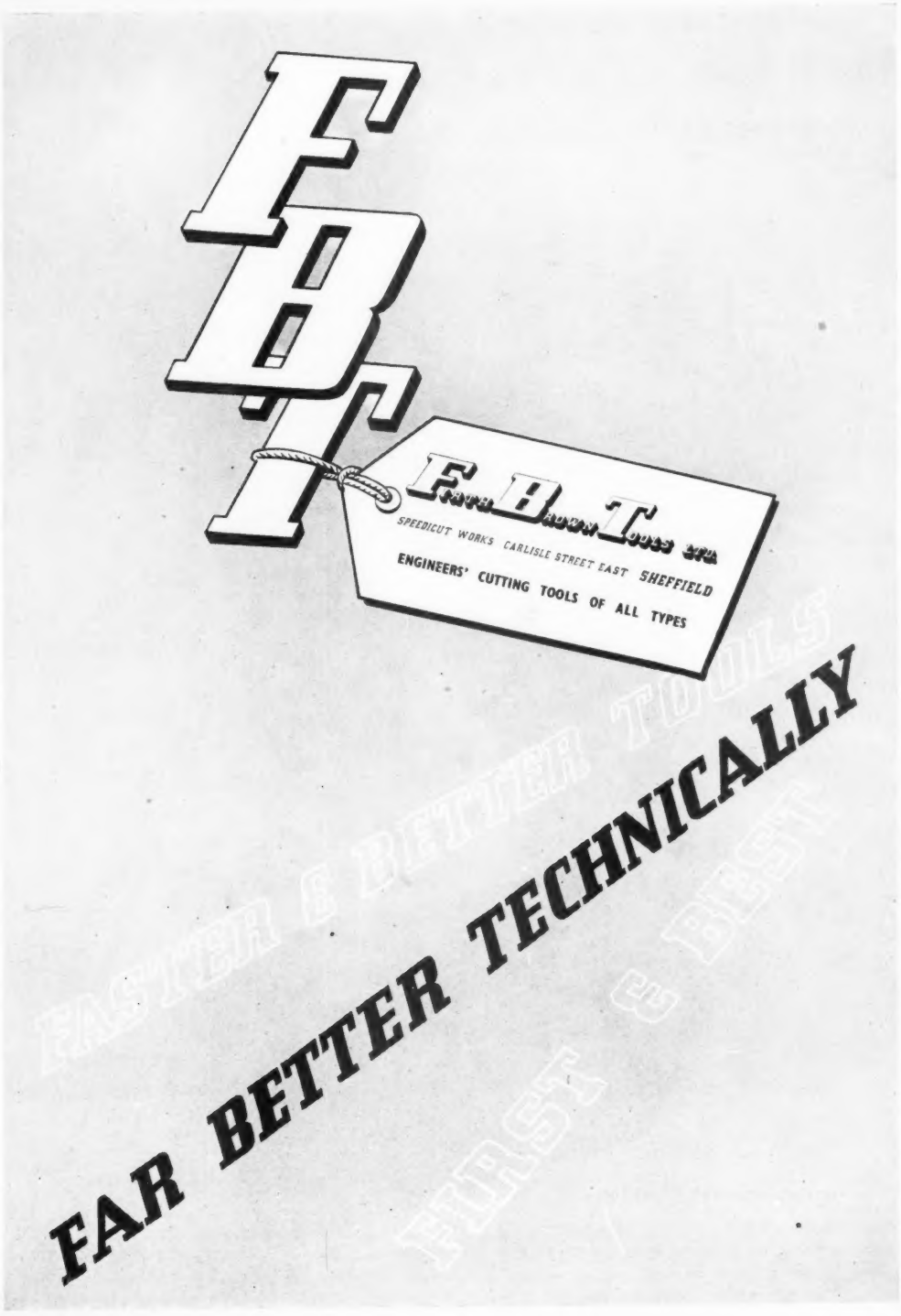
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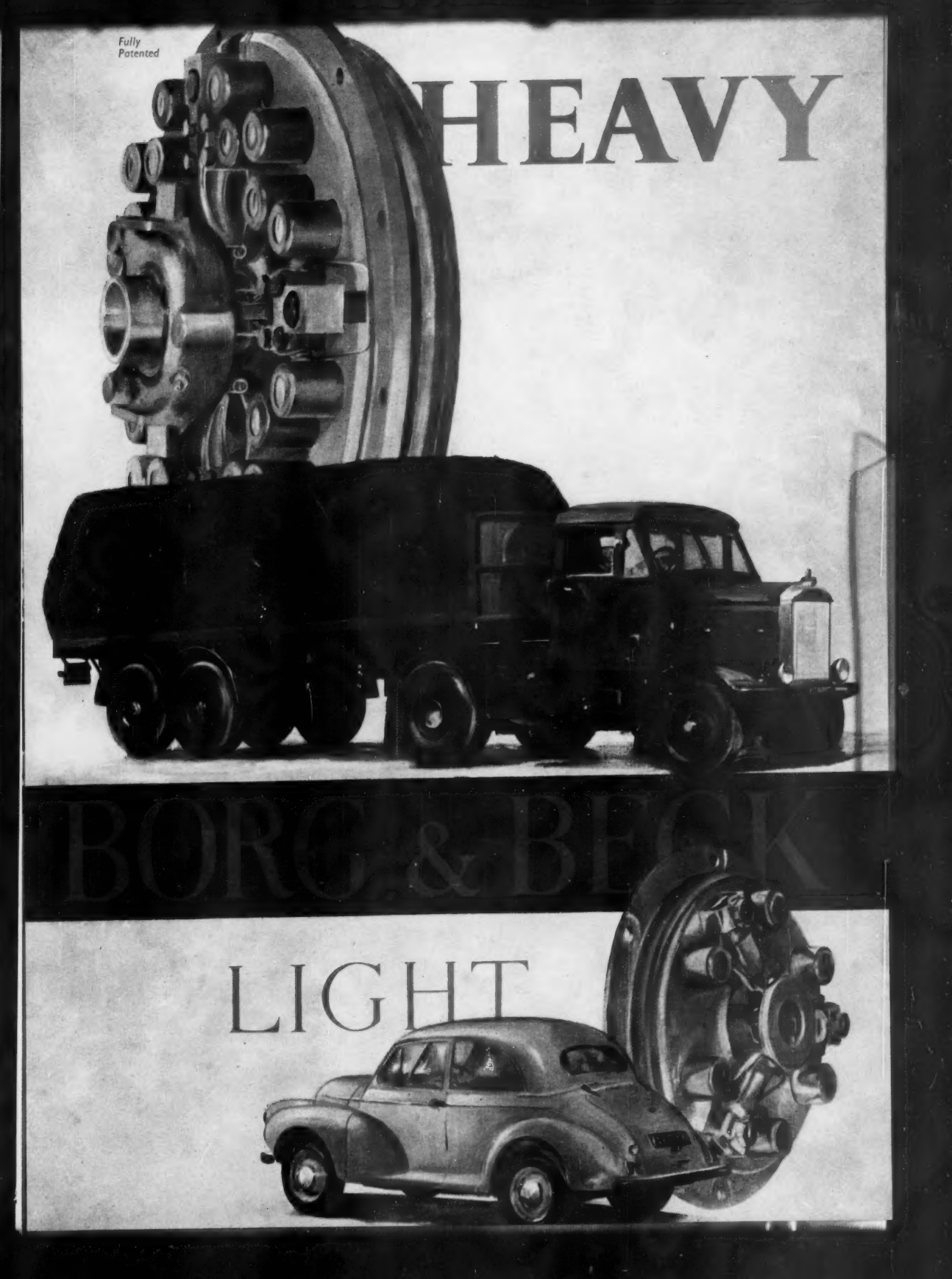
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that are solved by
Lockheed
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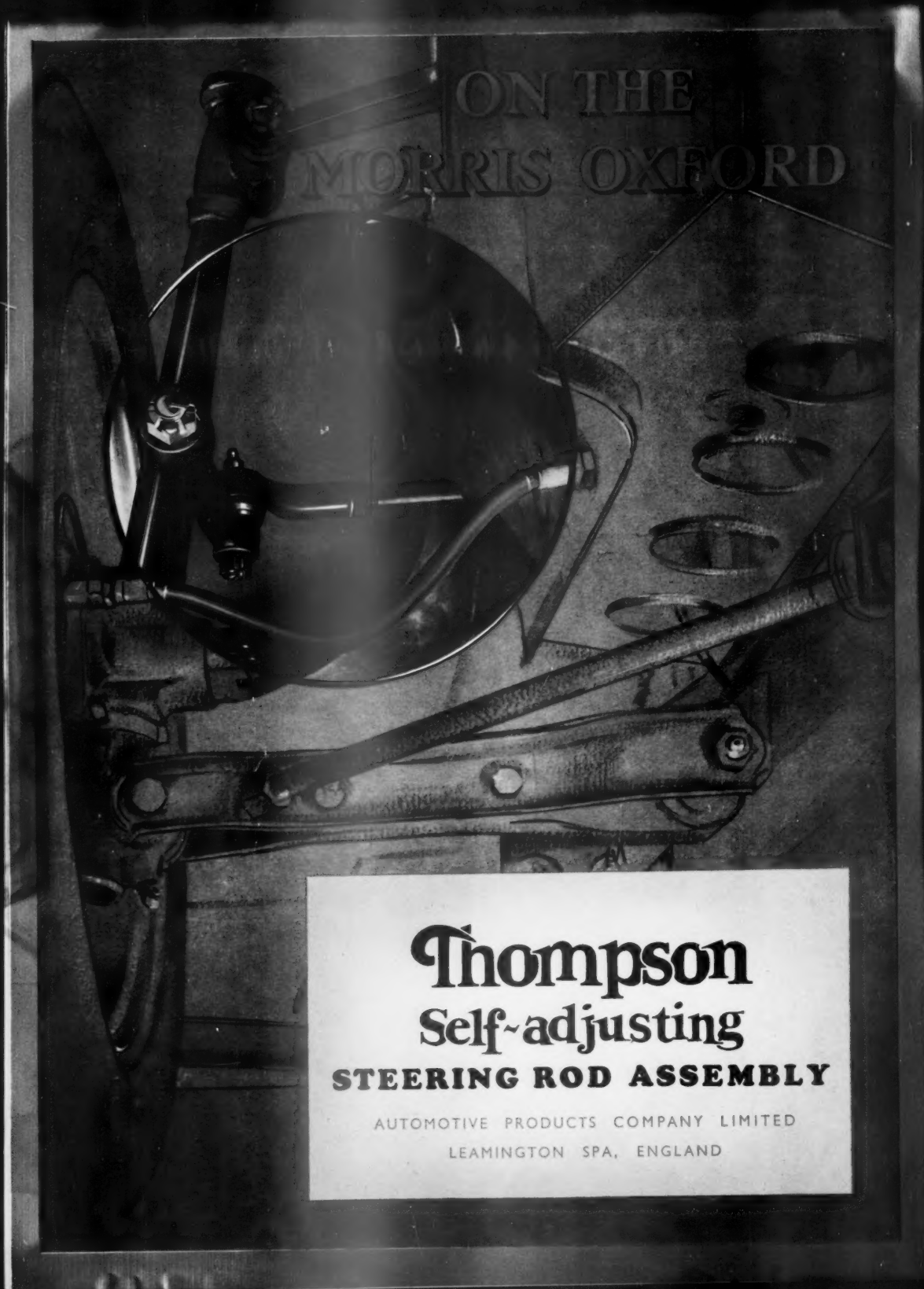

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Self-adjusting
STEERING ROD ASSEMBLY

AUTOMOTIVE PRODUCTS COMPANY LIMITED
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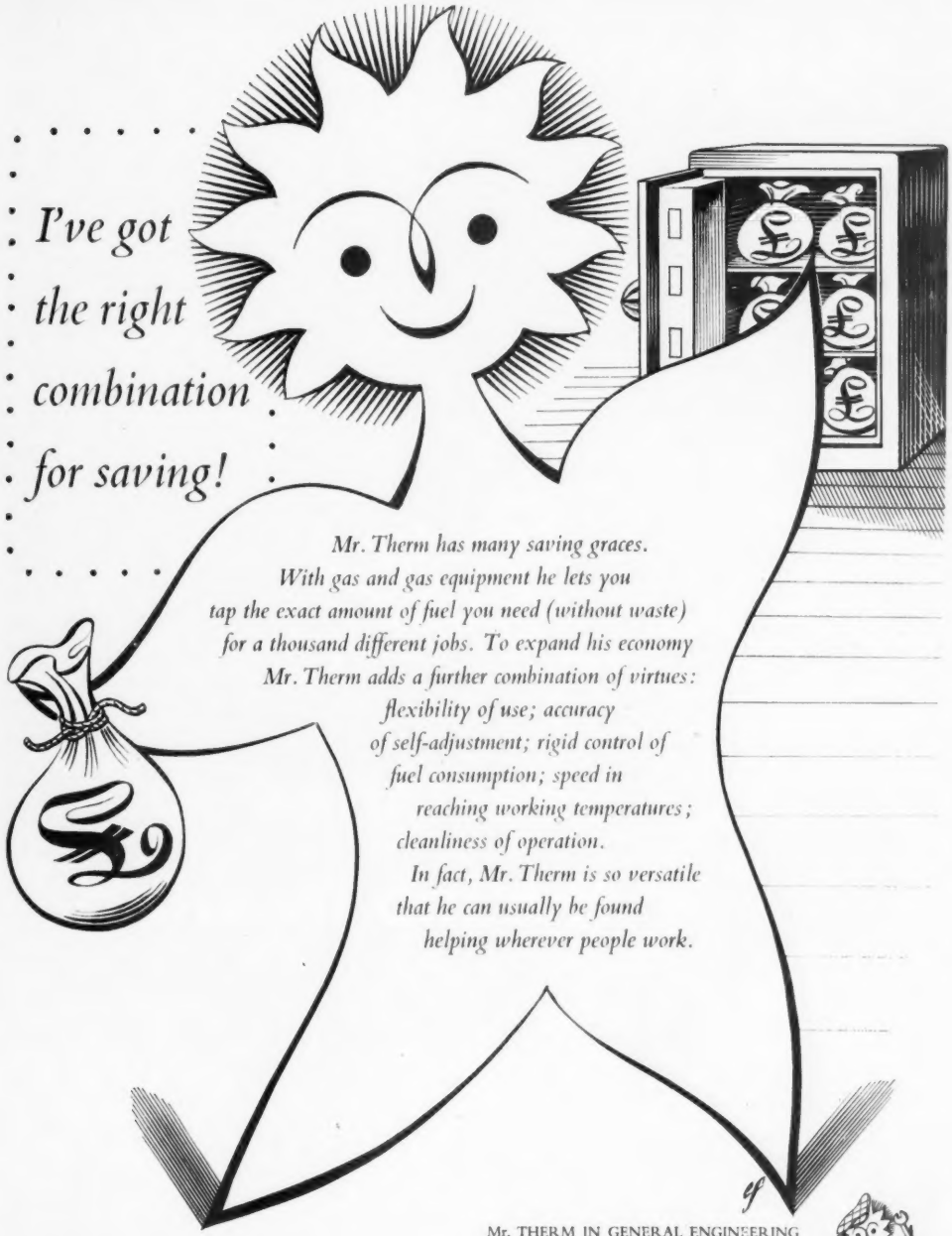

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Eclipse
TOOLS
designed with
the job in mind
Made by James Neill & Co. (Sheffield) Ltd. and obtainable from your usual

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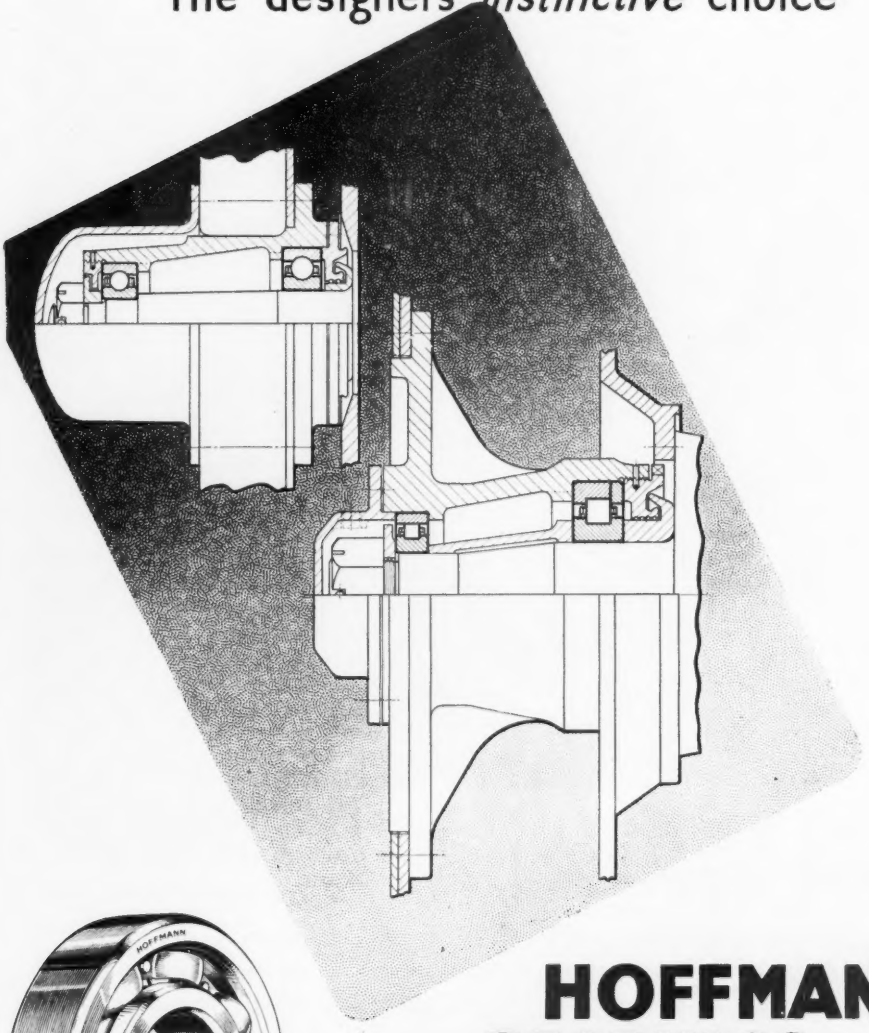
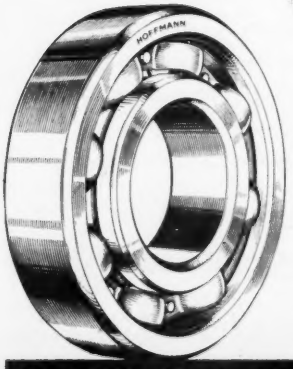
The designer's *instinctive* choice





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Front hubs must be free from slackness. Wear on the bearings causes shaky steering and wheel wobble resulting in uneven tyre wear. The wearlessness of Hoffmann Bearings is a well proved advantage.

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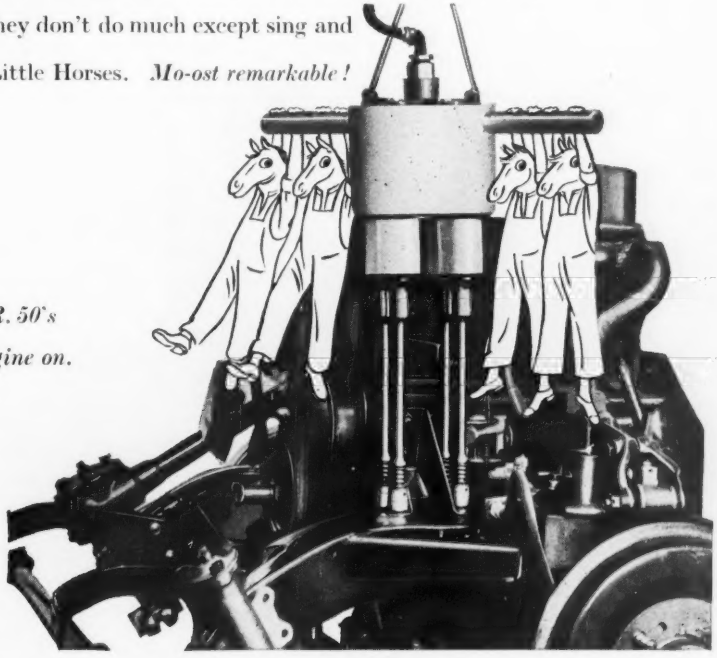
Four little horses depending on an AUSTIN

Mo-ost remarkable! — said the Managing Director, just returned from the new Austin Assembly Plant — they throw a handful of nuts and bolts on a moving platform and the next thing there's a fellow driving an A.40 off the other end. Of course it's my Little Horses who join the bits and pieces together. Little chaps called S.R.50 working singly or in teams of two, three or four. There are a few human beings about of course but they don't do much except sing and say "Hup!" to my Little Horses. *Mo-ost remarkable!*



Team of four S.R. 50's sticking the engine on.

Desoutter Power Tools Increase Production

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CRC232

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and tomorrow

At Shell's Thornton Research Centre in Cheshire—Britain's biggest scientific research centre for petroleum products—900 scientists and technicians work constantly to produce ever-better motor oils for your vehicles. After exhaustive laboratory experiments, the oils are subjected to gruelling practical engine tests *under actual running conditions*. Here you see a scientist operating the Two-Ball Machine—used for studying the performance of oils under conditions of extreme loading.

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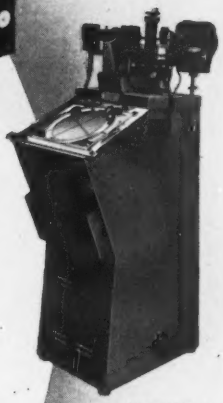

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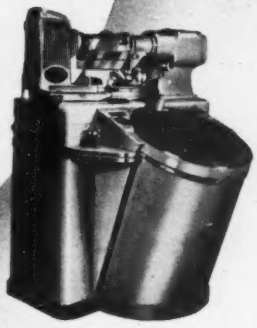

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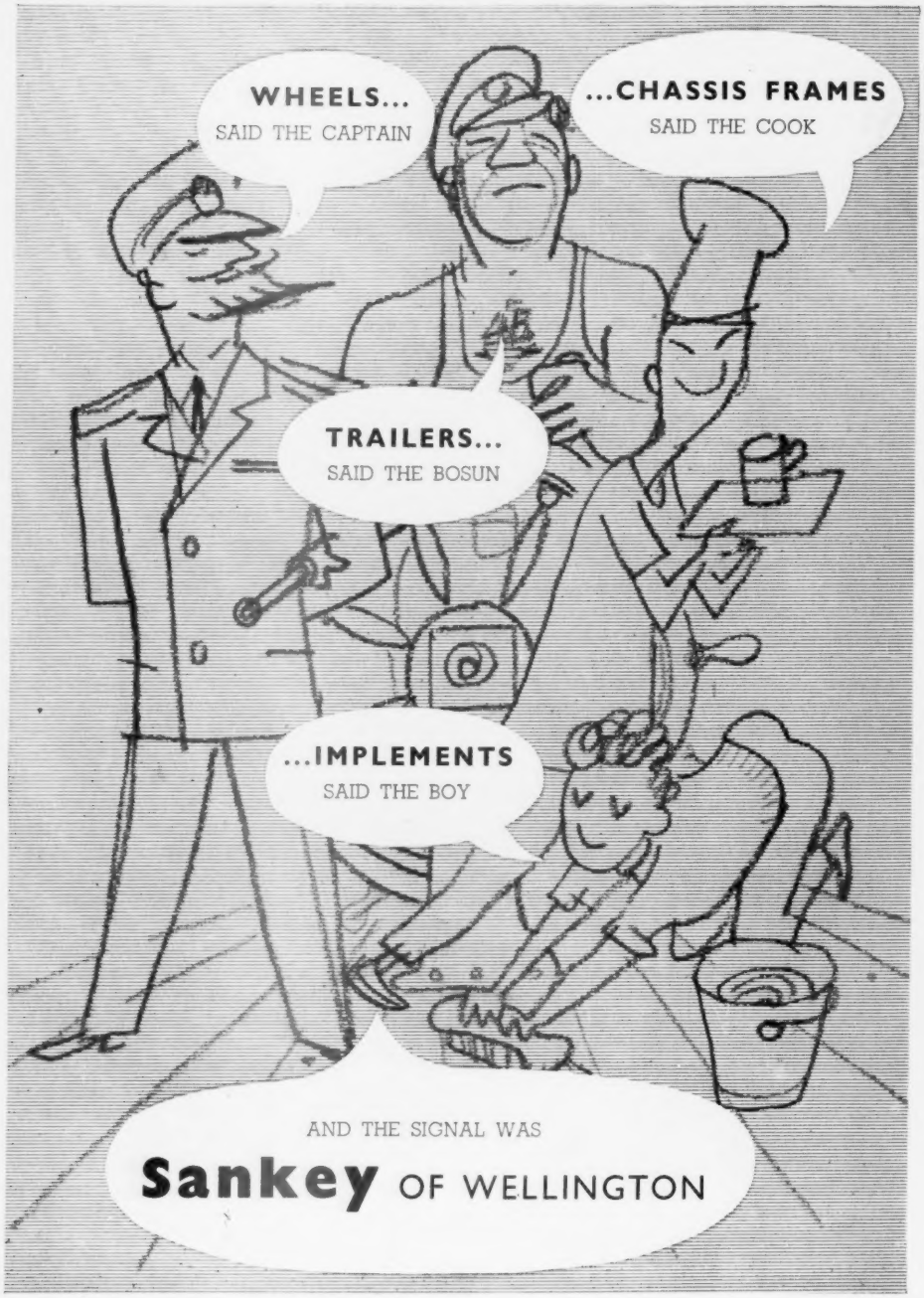

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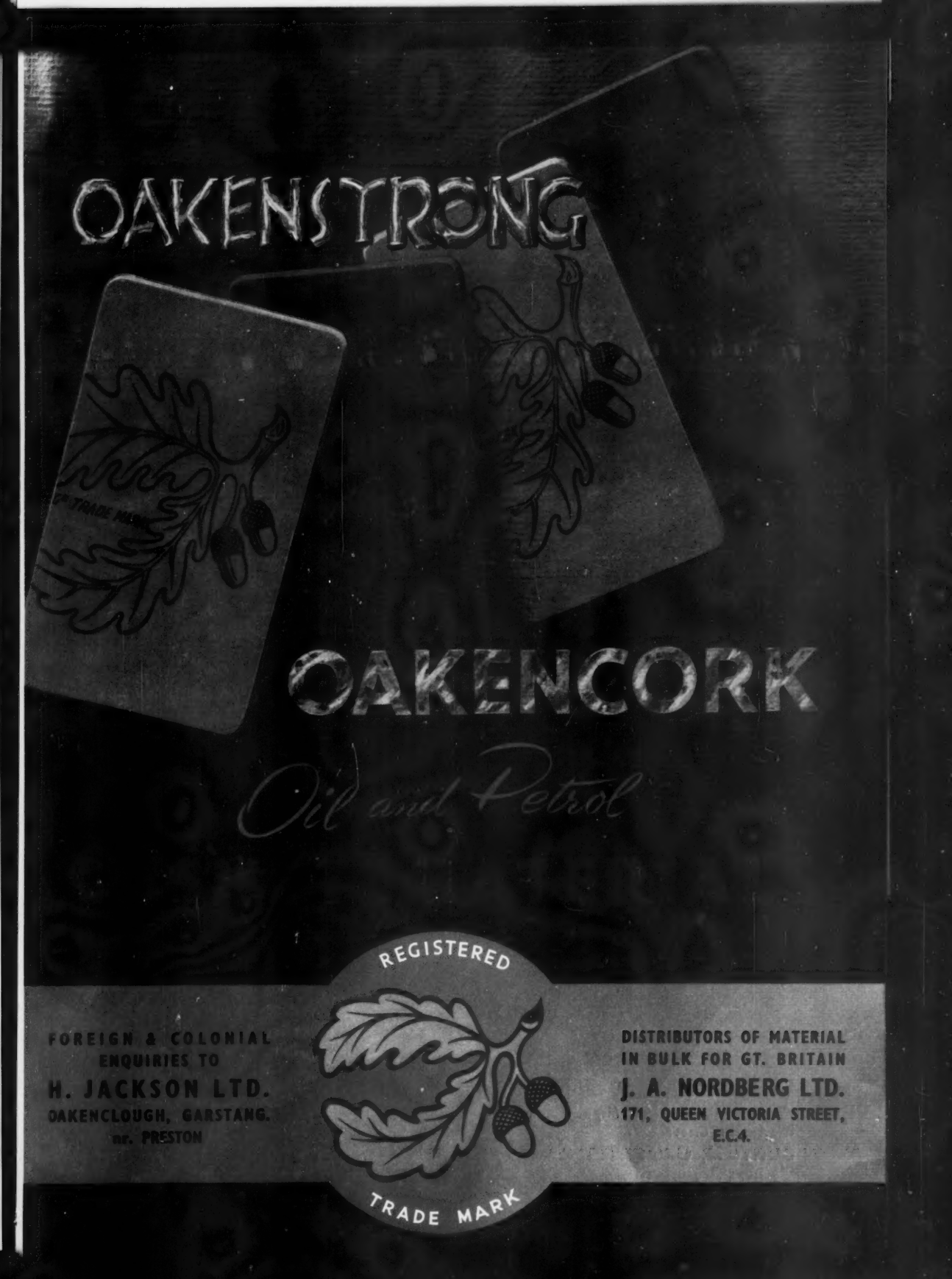
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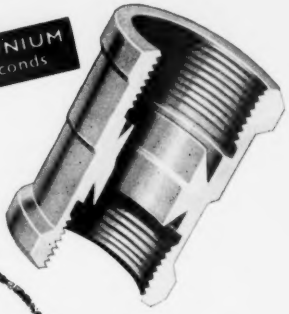
E.C.4.

18/8 STAINLESS STEEL
360 Seconds

BRASS
34 Seconds



ALUMINIUM
82 Seconds

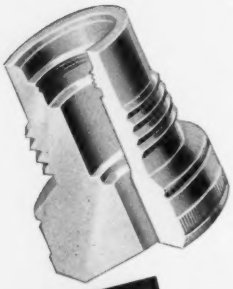
Typical Examples of work produced on
INDEX
Single-Spindle Automatics

BRASS
1·75 Seconds

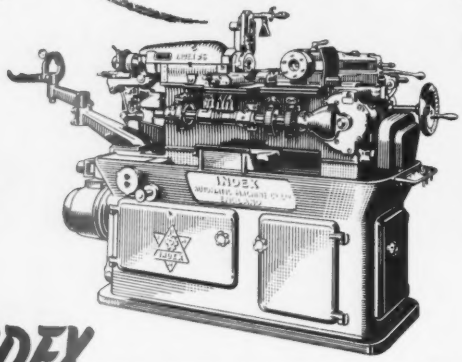
BRASS
3 Seconds

BRASS
6 Seconds

BRASS
1·75 Seconds



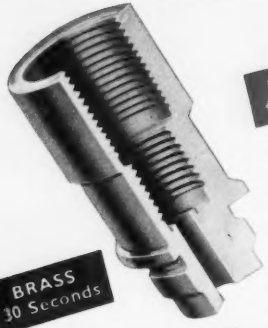
DURALUMIN
60 Seconds



See them in operation at Olympia

INDEX SINGLE - SPINDLE AUTOMATIC SCREW MACHINES
12, 24, 36, 52 m/m capacities.

I A
INDEX
Trade Mark



BRASS
30 Seconds

LEDLOY
48 Seconds

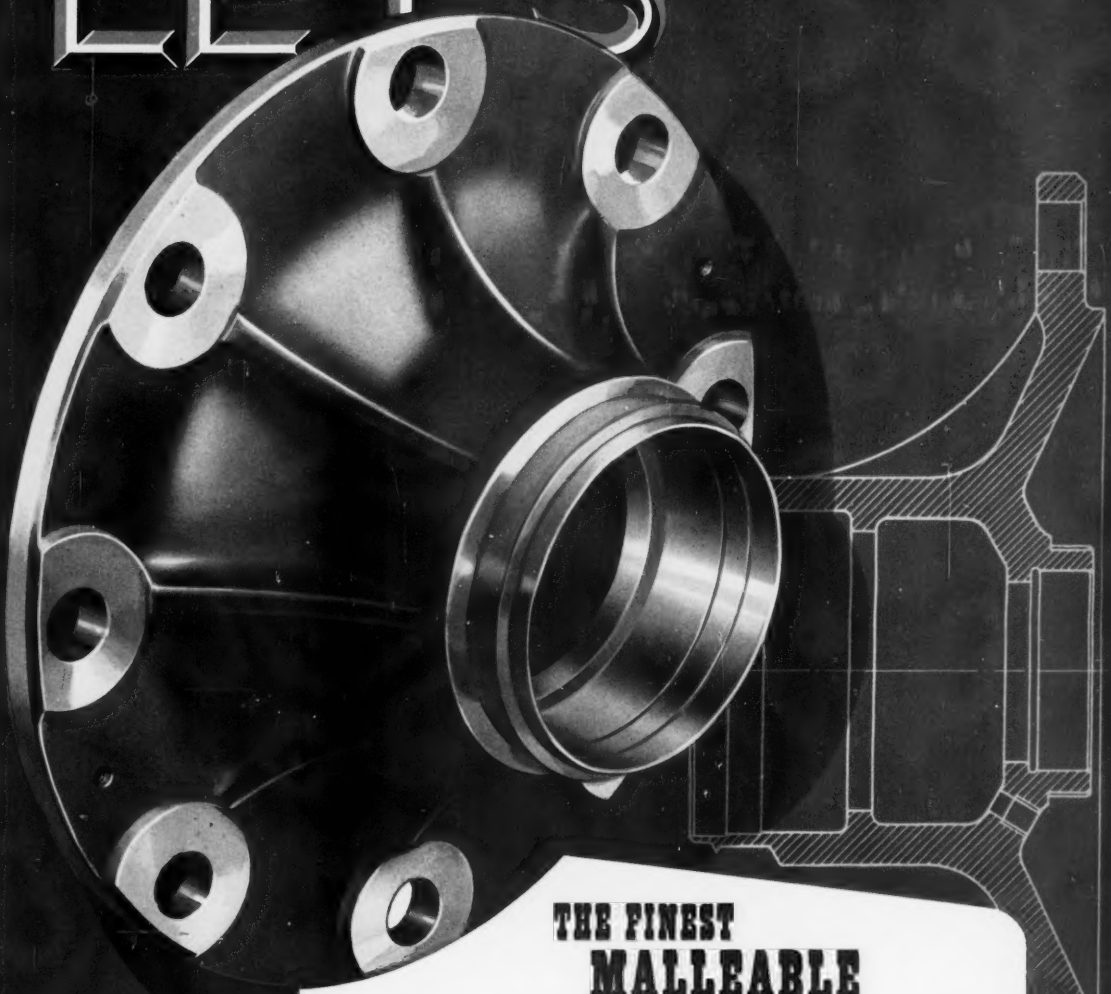
ALUMINIUM
36 Seconds

BRASS
56 Seconds

May we advise on the application of these machines to your work?

Sole Agents in Great Britain: (B.S.A. Tools Group)
BURTON, GRIFFITHS & CO., LTD., MARSTON GREEN, BIRMINGHAM

LEY'S



THE FINEST MALLEABLE

Ley's 'Black Heart' malleable castings are being used more and more for important parts where strength and rigidity must be combined with ease of machining. The illustration shows a front hub 12½" outside diameter, for a commercial vehicle.

LEY'S 'BLACK HEART' MALLEABLE CASTINGS:		LEY'S 'LEPAZ' MALLEABLE CASTINGS:	
Elongation:	18%	Elongation:	6%
Yield Point:	16 tons p.s.i.	Yield Point:	21 tons p.s.i.
Tensile Strength:	24 tons p.s.i.	Tensile Strength:	35 tons p.s.i.

We offer your designers the full collaboration of our engineers.

LEY'S MALLEABLE CASTINGS CO. LTD., DERBY. TELEPHONE: DERBY 45671

STRENGTH & RIGIDITY

LEY'S

Detachable oil filter

element easily inspected and replaced

This new AC partial-flow oil filter with replaceable element meets the increasing demand for a filter that can be removed for inspection and replaced every 8/10,000 miles. It provides a permanent installation for coupling into a by-pass oil supply system and the filtering element can be replaced without disturbing pipe connections. Sump oil is filtered on an average of ten times an hour. This means less motor wear and longer motor life — the objective of every motor manufacturer.

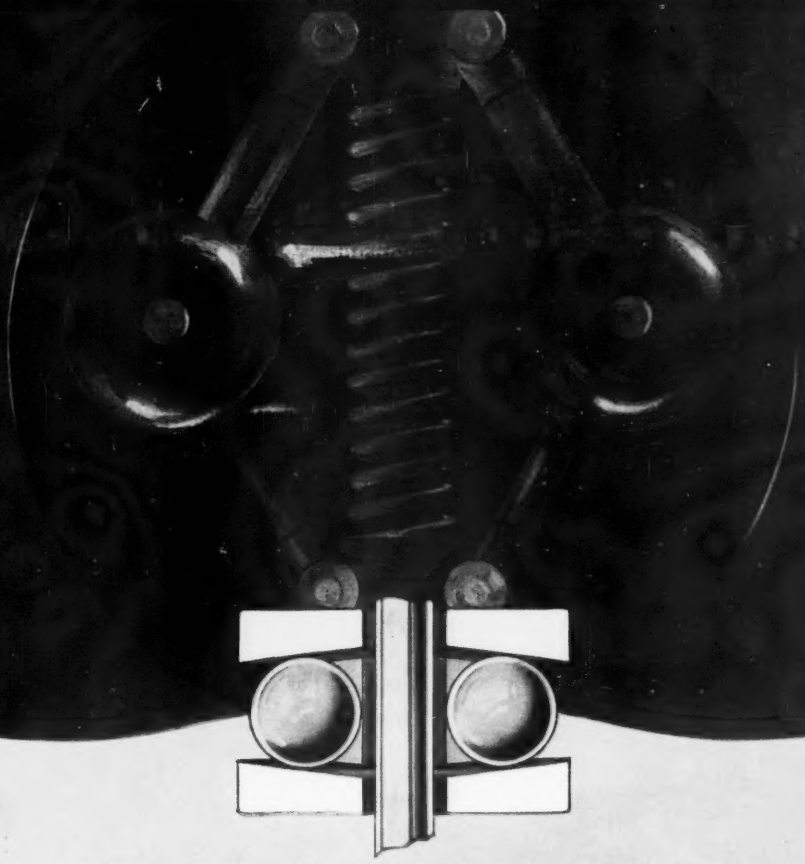
AC

AIR CLEANERS · AIR SILENCERS · CRANKCASE BREATHERS
CAR HEATERS · FUEL PUMPS · SPEEDOMETERS · GAUGES
INSTRUMENT PANELS · OIL FILTERS · THERMOSTATS
SPARK PLUGS · WINDSCREEN WIPERS · DIE CASTING

If you are contemplating an important new design, or a modification of an existing one, take advantage of the facilities of the AC Technical Bureau. We probably can save you hours of planning and experimentation. Write to the AC Technical Bureau, 54, The Butts, Coventry; or telephone Coventry 61747.

AC-SPHINX SPARK PLUG COMPANY,
DIVISION OF GENERAL MOTORS LTD.,
DUNSTABLE, BEDFORDSHIRE
and Southampton, Hants.

ACCURATE SPEED CONTROL



Iso-Speedic governors embody flybobs which consist of hardened steel balls rolling on ground tracks, eliminating all static friction, and resulting in a high degree of accuracy.

They are used on diesel engines and petrol engines, and are available for other speed control applications.

Iso-Speedic governors can be supplied which give control within 0.3% and are used on generators for radar and television. Other Iso-Speedic governors are available where a lower degree of accuracy is sufficient.

The services of our engineers are at your disposal.

Iso-Speedic

THE ISO-SPEEDIC COMPANY LTD., COVENTRY

SABOTAGE!

AT CHAPEL-EN-LE-FRITH

"Sabotage" in a loose sense is a term which may be applied to some of the experiments conducted in the Ferodo Test House as for example the testing to destruction of hundreds of Brake Linings weekly in routine check.

Another practical and valuable aspect of the Ferodo Physical Test Laboratory is to be found in the ingenious and specially designed apparatus which subjects samples to tests of endurance and efficiency far exceeding those encountered under the most strenuous road or racing conditions over a long period of time.

It is by such methods, backed by ceaseless chemical and physical research, that the insuperable standard of quality inherent in all Ferodo products is fully maintained.

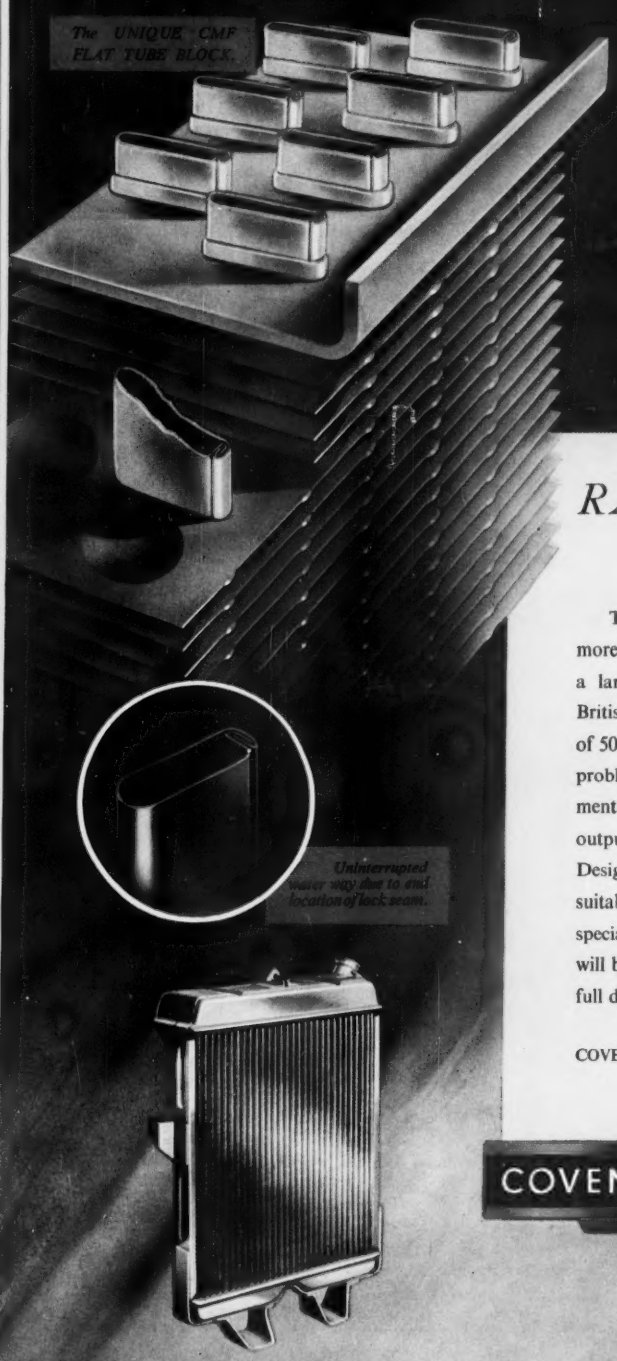
It is by close collaboration with Automobile Engineers and Designers, who at all times are freely and cordially invited to visit the Test House, that the problems associated with ever increasing h.p. and speed are dealt with . . . and solved.

FERODO

BRAKE & CLUTCH LININGS

FERODO LIMITED · CHAPEL-EN-LE-FRITH

A Member of the Turner & Newall Organisation



RADIATOR MAKER
to the Motor Trade

The Coventry Motor Fittings Co., Ltd., known more familiarly in the trade as CMF supplies annually a large quantity of the radiators used in leading British cars and commercial vehicles. Over a period of 50 years an unrivalled experience of cooling design problems has been built up, and the specialised equipment and production processes devised for large scale output of these exacting components are unique. Our Design Dept. is able at any time to put forward a suitable type of radiator or allied equipment for your special requirements. An enquiry to the address below will bring a qualified technical representative to take full details.

COVENTRY MOTOR FITTINGS CO. LTD., COVENTRY
Tel. Cov. 3144-5-6

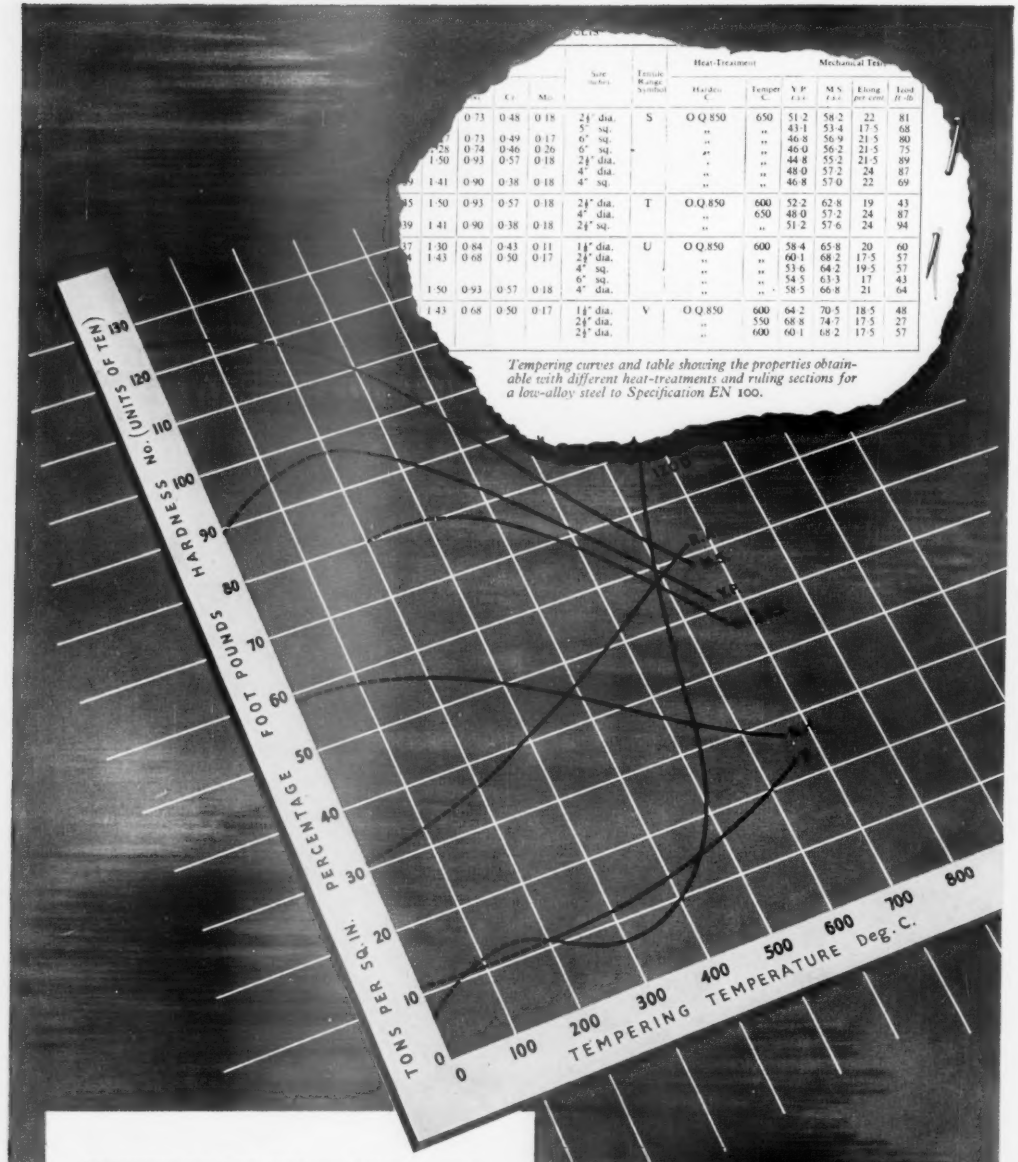
Cogent

COVENTRY MOTOR FITTINGS

CMF

			Size inches	Tensile Range Symbol	Heat-Treatment		Mechanical Tests			
	Cr.	Mo.			Hardens C.	Temper C.	Y.P. t.s.i.	M.S. t.s.i.	Elong. per cent	Izod ft.-lb.
0·73	0·48	0·18	2½" dia.	S	O.Q.850	650	51·2	58·2	22	81
			5" sq.		,,	,,	43·1	53·4	17·5	68
0·73	0·49	0·17	6" sq.		,,	,,	46·8	56·9	21·5	80
0·74	0·46	0·26	6" sq.		,,	,,	46·0	56·2	21·5	75
0·93	0·57	0·18	2½" dia.		,,	,,	44·8	55·2	21·5	89
			4" dia.		,,	,,	48·0	57·2	24	87
0·90	0·38	0·18	4" sq.		,,	,,	46·8	57·0	22	69
0·93	0·57	0·18	2½" dia.	T	O.Q.850	600	52·2	62·8	19	43
			4" dia.		,,	650	48·0	57·2	24	87
0·90	0·38	0·18	2½" sq.		,,	,,	51·2	57·6	24	94
0·84	0·43	0·11	1½" dia.	U	O.Q.850	600	58·4	65·8	20	60
0·68	0·50	0·17	2½" dia.		,,	,,	60·1	68·2	17·5	57
			4" sq.		,,	,,	53·6	64·2	19·5	57
			6" sq.		,,	,,	54·5	63·3	17	43
0·93	0·57	0·18	4" dia.		,,	,,	58·5	66·8	21	64
0·68	0·50	0·17	1½" dia.	V	O.Q.850	600	64·2	70·5	18·5	48
			2½" dia.		,,	550	68·8	74·7	17·5	27
			2½" dia.		,,	600	60·1	68·2	17·5	57

Tempering curves and table showing the properties obtainable with different heat-treatments and ruling sections for a low-alloy steel to Specification EN 100.



ECONOMY IN THE USE OF ALLOY STEELS

is a necessity today, and our publication "The Mechanical Properties of Nickel Alloy Steels" will assist the user to make two ends—properties and availability—meet. The tempering curve and table reproduced here are from this publication, which covers a wide range of direct-hardening, case-hardening and nitriding steels. Write for a free copy to:

THE MOND NICKEL COMPANY LIMITED

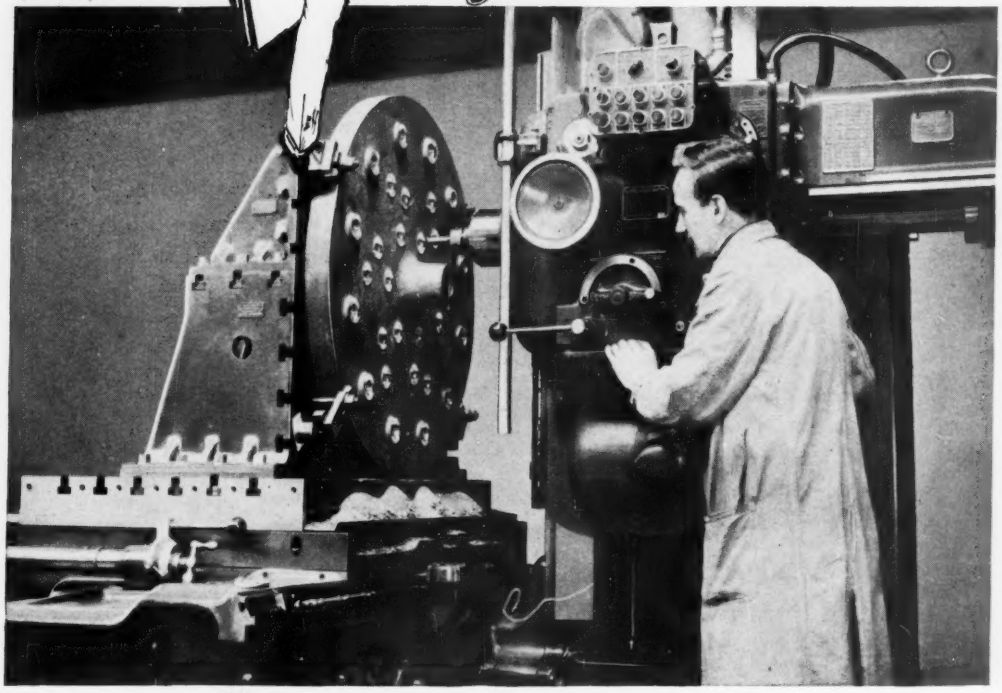
SUNDERLAND HOUSE · CURZON STREET · LONDON W1

12.SL.34

out of our ivory tower . . . .

Normally we are sufficient unto ourselves. We make the tools and many of the machine tools that make Holroyd Worm Gears, because this way we can be sure of living up to our very high standards of accuracy. But now and again we have to emerge from our ivory tower long enough to find a machine for a specific job — a machine we prefer not to make ourselves. We take any amount of trouble and put our hands deep into our pockets to secure the right one. Which attitude is why Holroyd Worm Gears are good enough for us to be boastful about.

The illustration shows one of the two De Vlieg Production Jig Boring Machines which form a part of our plant.



Holroyd WORM GEARS

JOHN HOLROYD & COMPANY LIMITED · MILNROW · LANCASHIRE

CRC 155

Fly away Peter

As easy as that? Of course not! The idea is born and developed, the plans are made and assembly is prepared. And this is where we come into the picture. We are there although you cannot see us. Bolts and Screws and Rivets are required and they must have a pedigree, just the same as the Hawker Hunter. The plane is built, it flies. Another fine aircraft has been helped on its way with our products, our knowledge and our resources. This we ensure by laboratory testing all raw materials and observing the closest metallurgical and process controls through every stage of manufacture of every fastener.

Linread LIMITED

We supply all industries requiring small fasteners of the highest quality to hold together the products they manufacture. Our specialists are always at your service to advise and assist you with your own particular problems.

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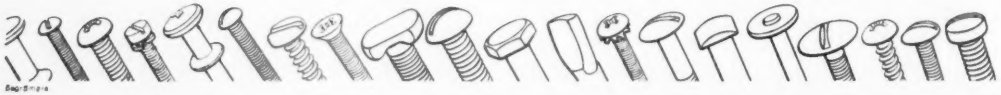

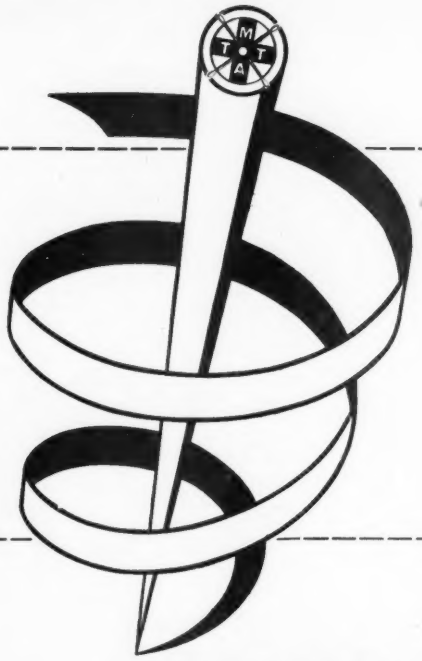

Here is an exceptional opportunity for you to inspect and compare the finest machine tools and allied products from the leading manufacturing countries of the world.

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OLYMPIA · LONDON · SEPT. 17 - OCT. 4

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THE LESS THE MAN AT THE BENCH MOVES ABOUT, the greater his output will be. It pays to have him standing still. If you cannot reduce his unproductive movements by changing your factory layout, you may still be able to do so by installing electrically driven mechanical handling equipment. And by giving him electrically powered hand tools you will put both the power and the work directly into his hands. By using electricity in these ways you can raise productivity and *make efficient use of the available supplies of power.*

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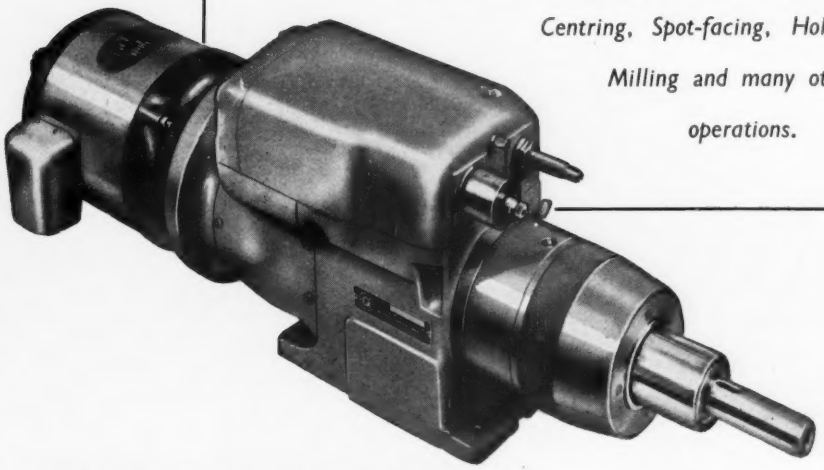
Controlled power for Drilling, Reaming, Centring, Spot-facing, Hollow Milling and many other operations.



QUICK SET-UP, HYDRAULIC FEED CONTROL
FAST TRAVERSE on approach and return
SIMPLE, COMPACT, RUGGED DESIGN
UNIVERSAL MOUNTING.

Any combination of these heads can be rapidly built up to suit any production problem.

The units can be supplied for direct drive, gear reduction drive, or for pulley drive.

Available in two sizes: Capacities $\frac{5}{16}$" and $\frac{1}{2}$" dia. drill.

Sole distributors for Delta-Milwaukee and Tauco.

We will supply the heads or build any special machine incorporating these units.

● *Early delivery*

GASTON E. MARBAIX LTD

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This special ARCHDALE multi-head milling machine helps to speed production on exhaust manifolds at Vauxhall Motors Limited, Luton.
Developing such machines is a speciality of the ARCHDALE organisation, and the machine illustrated is only one of hundreds of special drilling, tapping, reaming, boring and milling machines supplied to the automobile industry.
Our wide experience is at your service and we shall be pleased to co-operate with you on your own production problems.
MACHINERY

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SOLE AGENTS . ALFRED HERBERT LTD., COVENTRY

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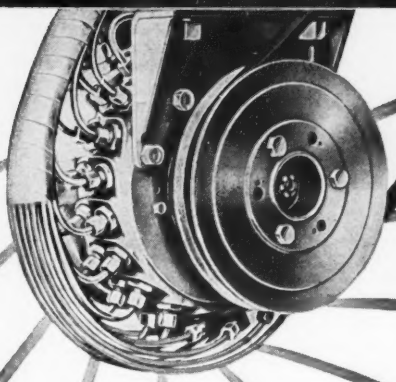
BUSES & COMMERCIAL VEHICLES

by means of

'Syndromic'

MILEAGE CONTROLLED LUBRICATOR



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Backed by Tecalemit design and service.

Pre-determined quantities of oil to each point.

Adequate and consistent lubrication ensured throughout.

Eminently suited to fleet conversion.

The outcome of over 20 years experience.

All 40 main points lubricated individually.

Self-lubricated pump requiring no attention.

Oil fed under high pressure

Lubrication carried out on a definite mileage basis.

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The Authority on Lubrication

PLYMOUTH, ENGLAND

T300

. . . *Cornelly Quarry* . . .

THIS highly mechanized Limestone Quarry serving The Steel Company of Wales is situated about six miles from the Abbey Works, Port Talbot. It is equipped with electric well-boring machines, $3\frac{1}{2}$ and $2\frac{1}{2}$ cubic yard electric shovels, 12-ton diesel lorries and primary, secondary and tertiary crushers. The Quarry is able to cope with substantial demands for large development schemes etc. Keen prices quoted and immediate delivery possible by road or rail in all sizes, viz :—

Sized Limestone from 12″ to 4″

Concrete Aggregate in all sizes from 2″ to $\frac{3}{8}$″

Limestone Dust and Stone Filling

THE STEEL COMPANY OF WALES LIMITED

ABBEY WORKS · PORT TALBOT

EACH TO ITS OWN JOB!

DORMER

ENGINEERS' SMALL TOOLS

for consistency

STAND No. 226

THE SHEFFIELD TWIST DRILL AND STEEL COMPANY LIMITED

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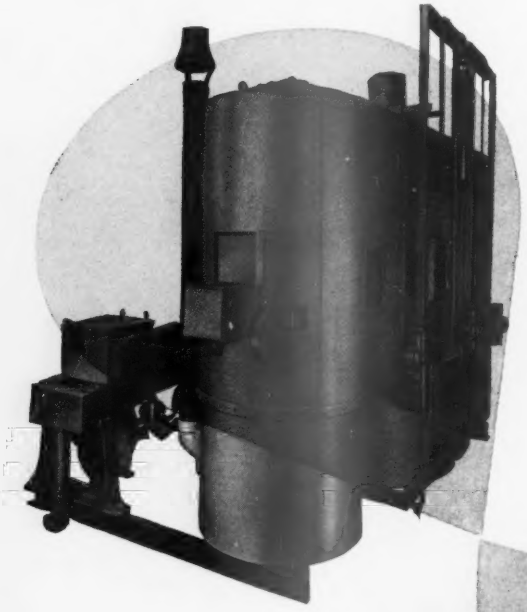
DORMER TOOLS ARE OBTAINABLE FROM YOUR USUAL ENGINEERS' MERCHANTS

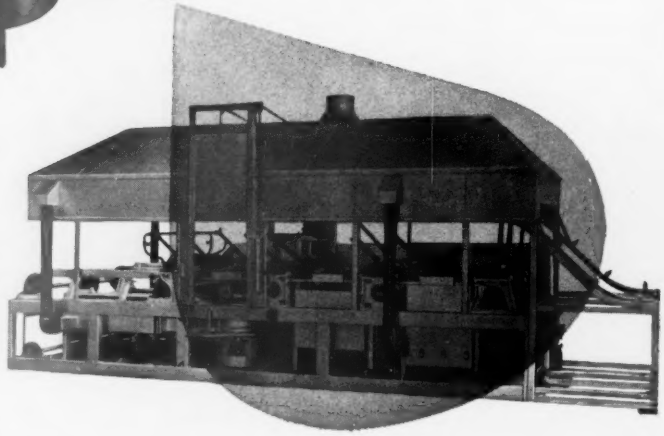
'Cassel' Mechanised Salt Bath Furnaces

for Carburising, Heat-treatment, Austempering, Martempering and Brazing.



All the advantages of the salt bath process plus

- **Flow production at high rates**
- **Less handling and lower labour costs**
- **Uniform results**
- **Automatic control of time and temperature**
- **Elimination of fume and dust**



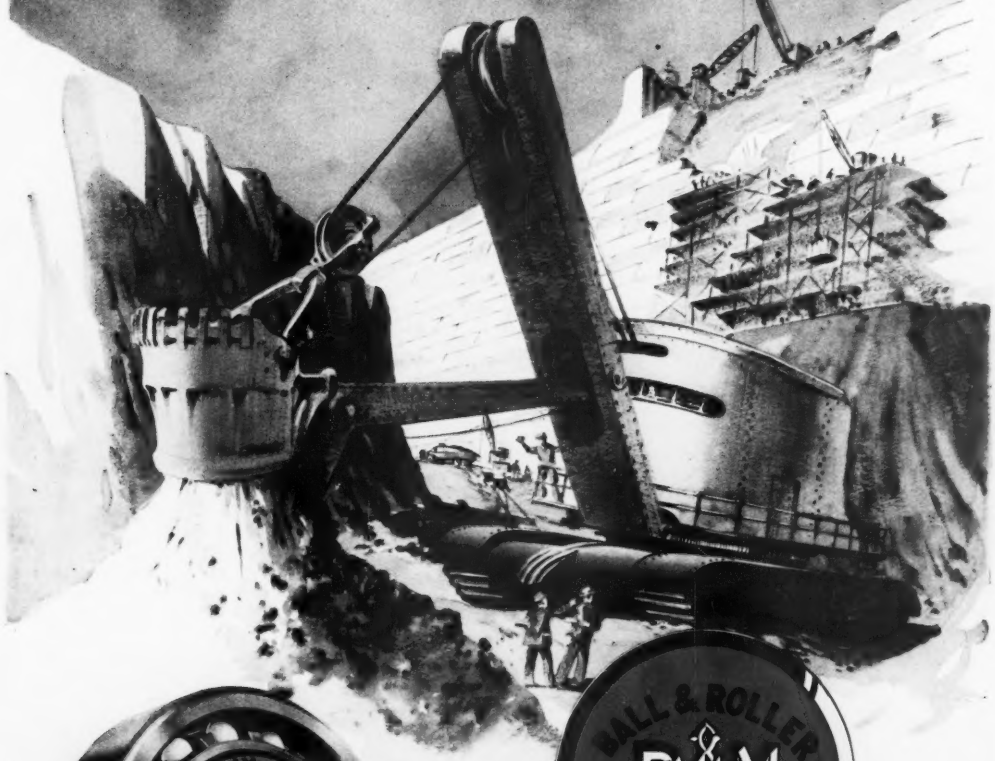
I.C.I. design and manufacture mechanised salt bath furnaces to meet individual requirements. For further information, or for details of 'Cassel' salts and standard types of furnace, consult—

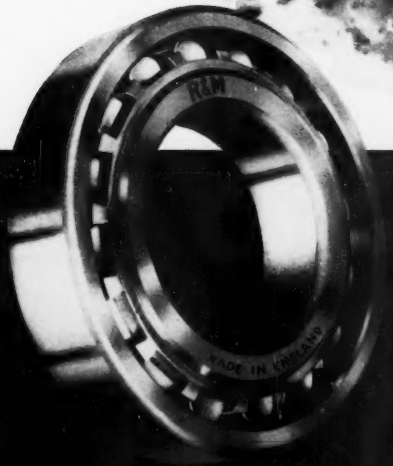
ICI

IMPERIAL CHEMICAL INDUSTRIES LIMITED, LONDON, S.W.1

C.C.156

AHEAD OF ACHIEVEMENT





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MECHANICAL EARTH MOVERS.

We are prepared for the day when Building projects will be top priority. Excavators, Bulldozers, etc., perform the heaviest, most gruelling tasks with the utmost dependability, while R&M Bearings carry the loads.

If you require anti-friction Bearings for tomorrow's designs, you will assuredly find a Bearing to suit your requirements among the R&M Comprehensive Range.

RANSOME & MARLES BEARING CO. LTD., NEWARK-ON-TRENT, ENGLAND

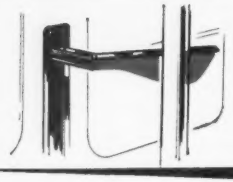
K/RM 87

C.A.V.

'T' TYPE DIRECTION INDICATOR

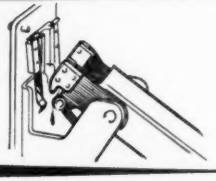
with the unique FLEXING ARM

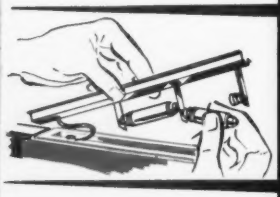
Specially designed for Commercial Vehicles



Spring controlled flexing arm prevents damage to mechanism if an obstacle is struck accidentally.

Hinge switch eliminates flexible leads to arm. Indicator lamps operate only in fully 'Up' position.





Twin bulbs in parallel as safeguard against failure, spring mounted to minimise shock. Bulb unit readily accessible.

Delay switch cancels signal automatically after pre-set period (adjustable 5 to 20 secs.). Man-size switch for operation by gloved hands, with illuminated front, lit only when arm is fully raised.

C.A.V.

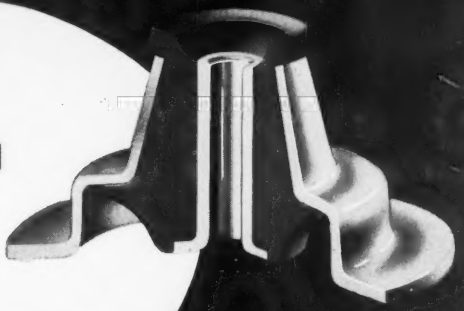
Fuel Injection and Electrical Equipment

C. A. V. LIMITED. ACTON. LONDON. W. 3

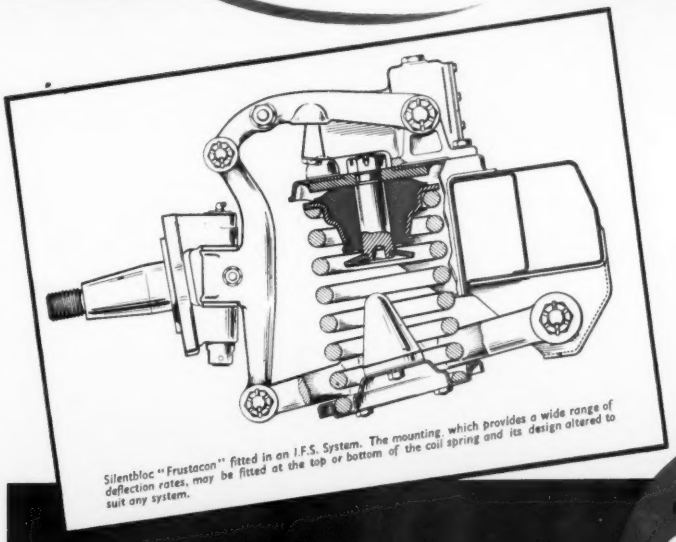
174-465

Silentbloc FRUSTACON..

. . . the first scientifically designed flexible mounting fitted to the I.F.S. of a Production Car



The 'Frustacon' mounting fitted to the I.F.S. of the 1952 Rover '75'. Sectioned to show rubber.



Silentbloc "Frustacon" fitted in an I.F.S. System. The mounting, which provides a wide range of deflection rates, may be fitted at the top or bottom of the coil spring and its design altered to suit any system.

To minimise the transmission of road noise to the body by the coil springs, Silentbloc Frustacon mountings have been incorporated in the I.F.S. of the 1952 Rover '75'. This notable step forward in the reduction of noise is another example of co-operation between car manufacturers and Silentbloc Ltd.

DESIGNERS AND MANUFACTURERS OF RESILIENT UNITS FOR MORE THAN 25 YEARS

SILENTBLOC LTD.

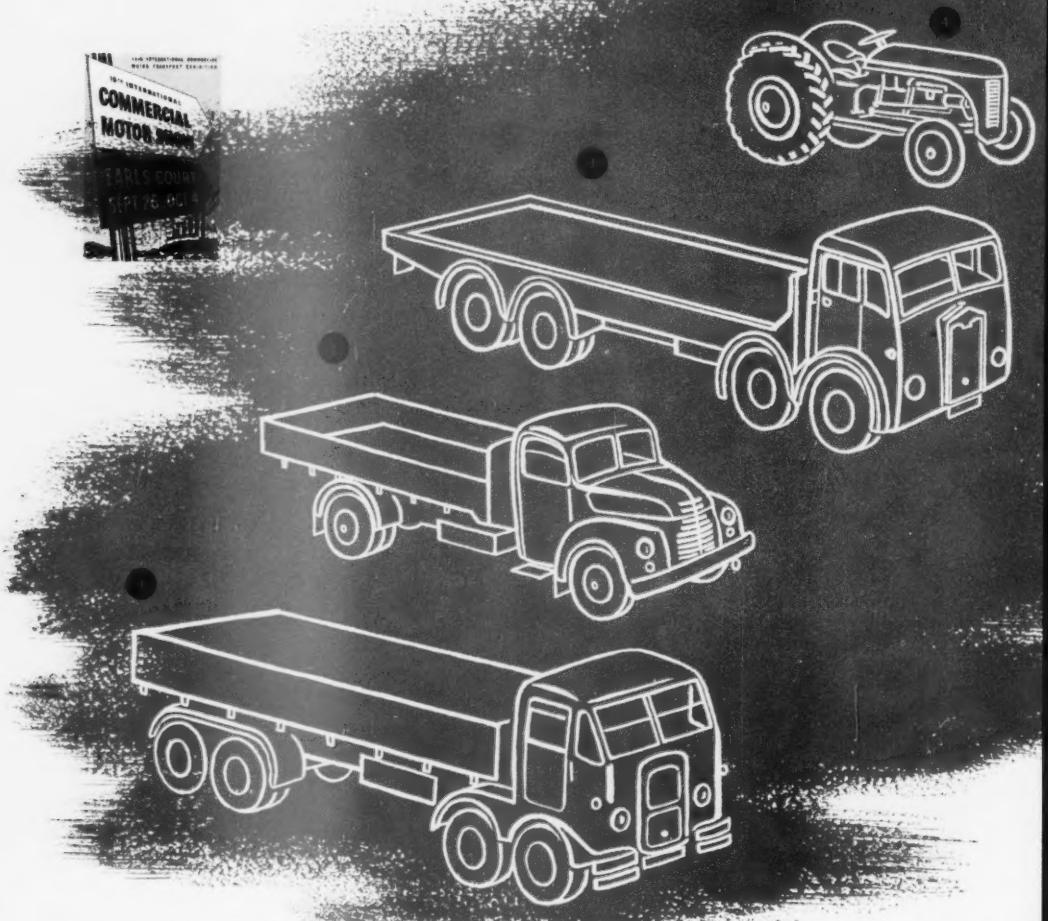
VICTORIA GARDENS

LONDON, W. II. TEL. PARK 9821

FIRST!.. AND

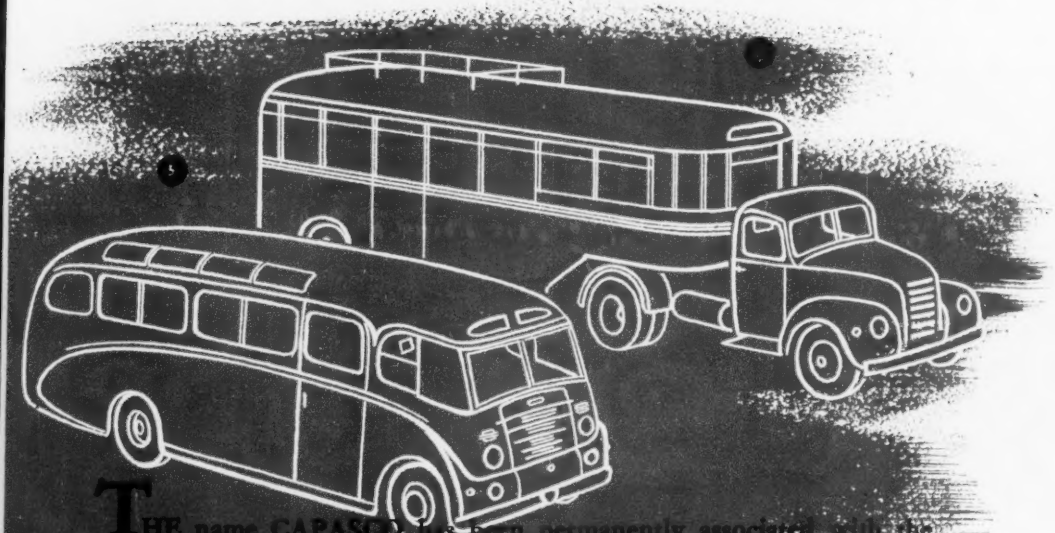
CAPASCO

NON-FADE *Moulded* BRAKE LININGS



1 Scammell 15-ton Rigid 8-Wheeler. **2** Leyland 'Comet 90' Truck Model ECO2.1R. **3** Albion 8-Wheeled Heavy Duty Vehicle. **4** Ferguson Standard Tractor. **5** Tilling-Stevens Express Type. L4MA8. **6** British Trailer Company's Airflow II 50-Seat Semitrailer Coach.

STILL FOREMOST



THE name CAPASCO has been permanently associated with the development and perfection of Heavy Duty Moulded Brake Linings right from the start.

CAPASCO pioneered the way, and is still leading this highly specialised field. CAPASCO Moulded Brake Linings are daily proving their efficiency and reliability under the most arduous conditions and in all forms of Transport.

The illustrations demonstrate clearly a few of the extensive range of vehicles to which CAPASCO is now fitted as original equipment.

★ Press Moulded Homogeneous Structure ★ High Mechanical and Impact Strength ★ Rapid 'Set' Recovery ★ Extreme Resistance to Frictional Fade ★ Dimensional Stability ★ Uniform Wear to 'Wafer' Thickness ★ Impervious to Oil and Grease ★ Non-abrasive to Brake Drums ★ Suitable for Medium and Heavy Duty Application involving High Temperatures

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The final choice of material for high-temperature plant is not easy. That is where we can help. We have service results covering materials used in widely differing conditions, and it is possible that we already have the answer to your problem. The chances are that one of the range of high-nickel alloys will be the most suitable—but ask us *before* you go ahead.

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"It has given us such excellent service . . . . let us have catalogues of your full range of machines because it will take very high pressure sales talk to sell us another make if a suitable one is in your range."

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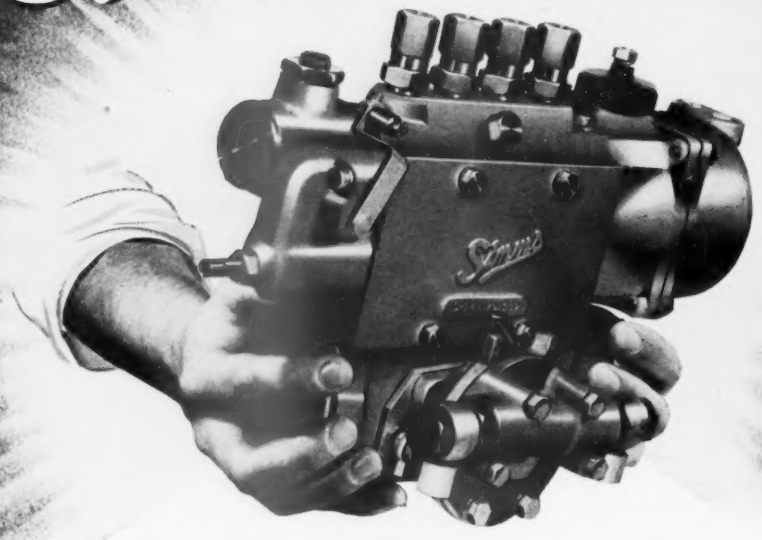

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NEW... elements, highly resistant to wear from dirty or badly-filtered fuel.

NEW... method of adjustment simplifies calibration and reduces backlash.

NEW... large diameter stiff camshaft to withstand higher pressures.

NEW... method of pneumatic governing, giving consistant performance over the whole speed range with particularly steady idling and closely controlled run out.

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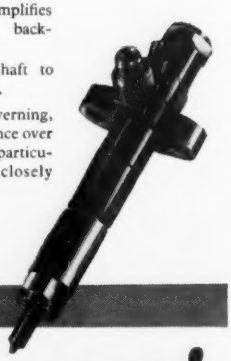

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DURALUMIN *body*

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450 cu. ft. capacity - U.W. 2½ tons

This combined box van, built on a Bedford 3-4 ton chassis, is just another example of the advantages obtained by using 'Duralumin' in the construction of commercial motor bodies. It has a full loading capacity of 13 ft. 4 ins. x 6 ft. x 5 ft. 8 ins., although the body and cab together weigh only three-quarters of a ton and the entire vehicle weighs just over 2½ tons unladen.

'Duralumin' bodies are tough and durable, remaining weather-proof for long periods with the minimum of attention. And because 'Duralumin' is only a third the weight of steel, the bodies can be built much more robustly, without unduly increasing the weight. Tyre wear and fuel consumption are reduced, and in many cases, such as that of the van illustrated, the use of 'Duralumin' enables the unladen weight to be kept below the 3-ton level, giving a considerable saving in the rate of tax paid.

'Duralumin' was developed and is manufactured exclusively by James Booth & Company Limited. This photograph is published by courtesy of the Duramin Engineering Company Limited of Park Royal, London, N.W.10 — a wholly independent company which specifies 'Duralumin' because, like many other coach and body builders, it has proved it to be the most reliable of light aluminium alloys.

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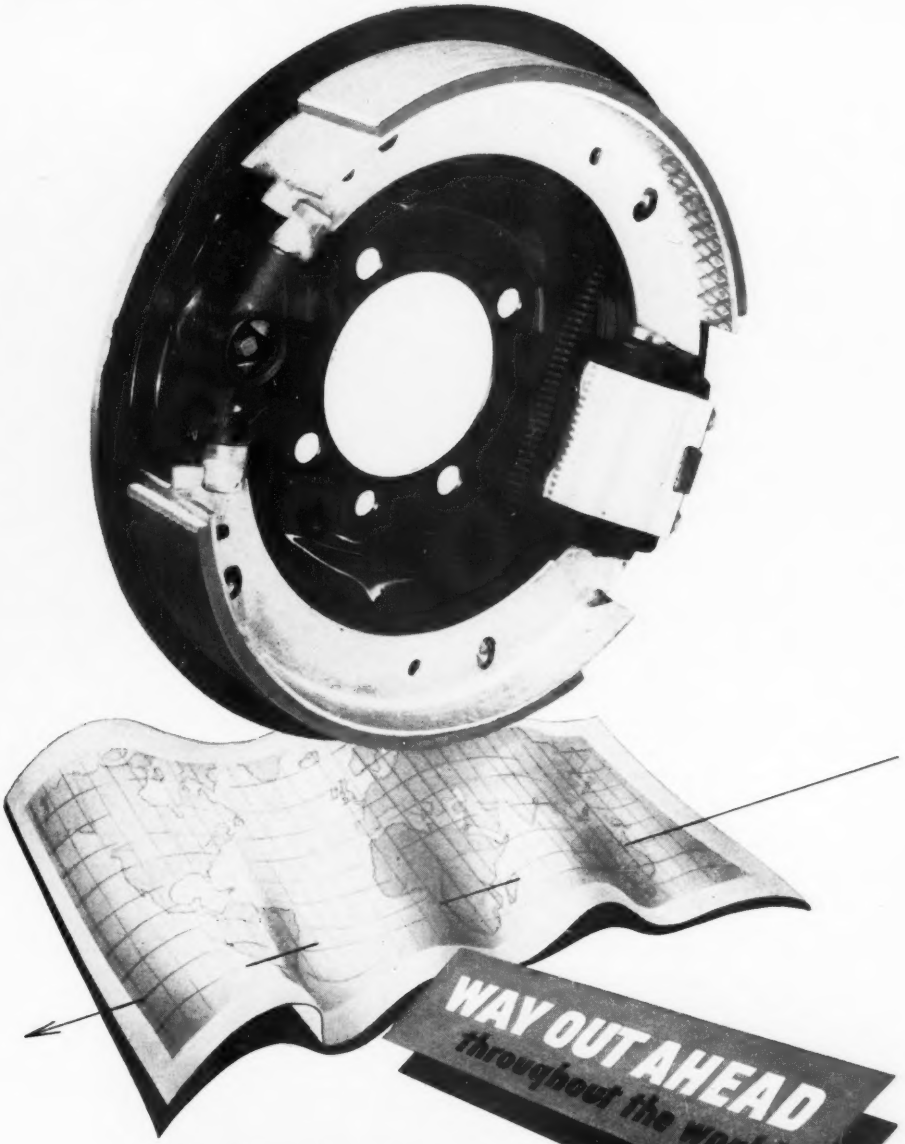

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EATON

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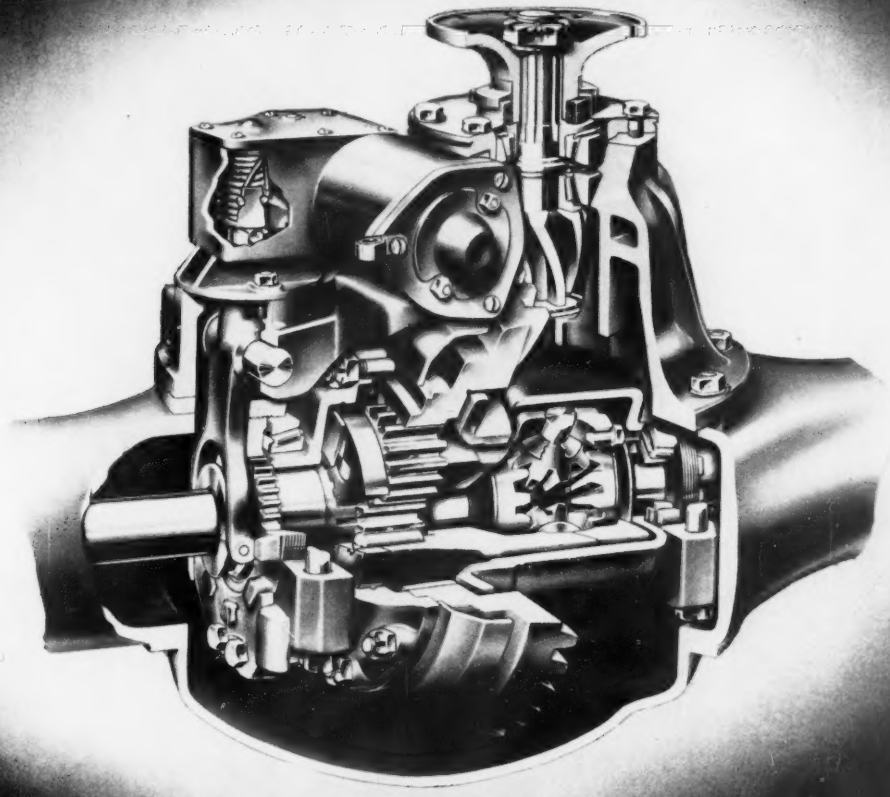

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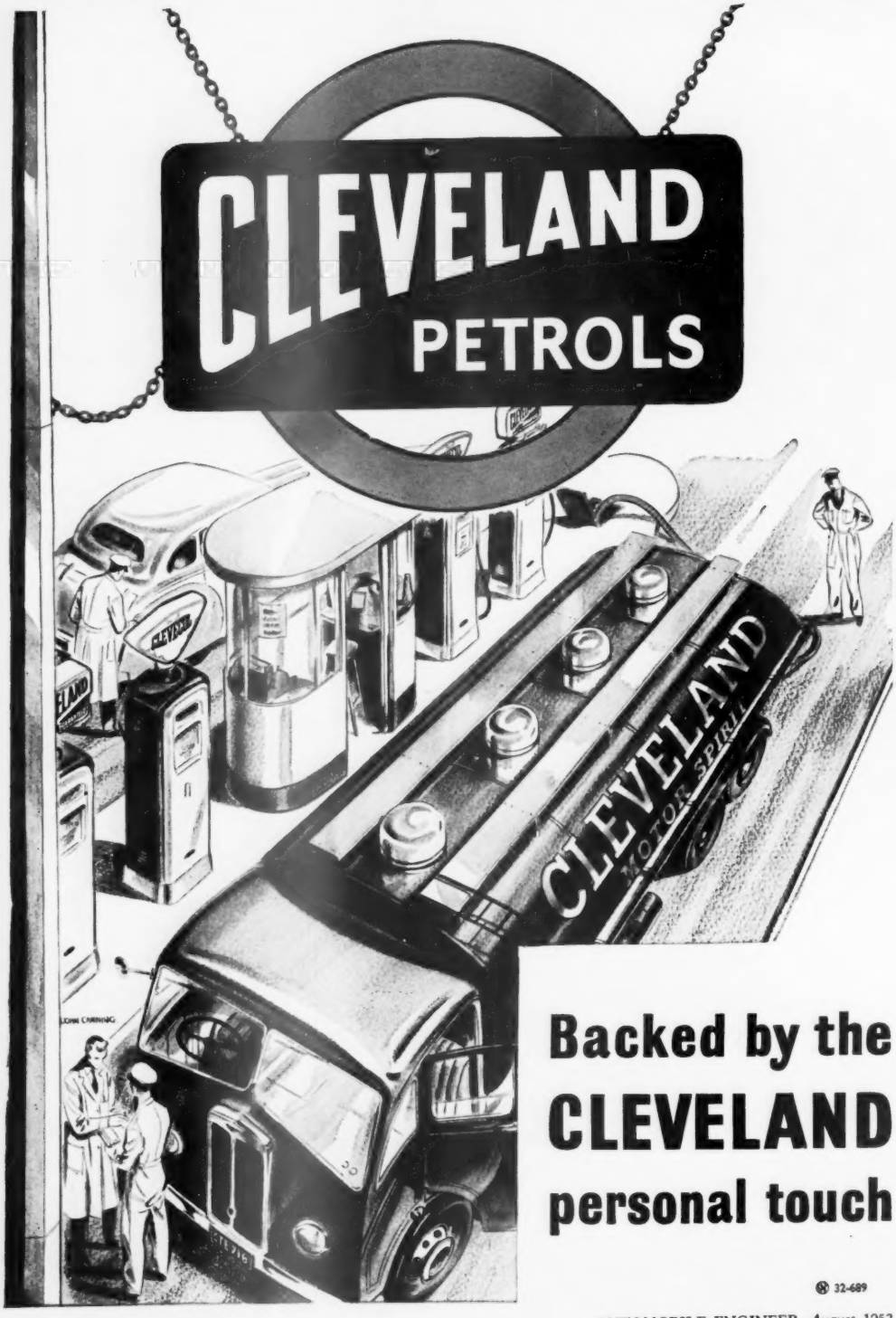

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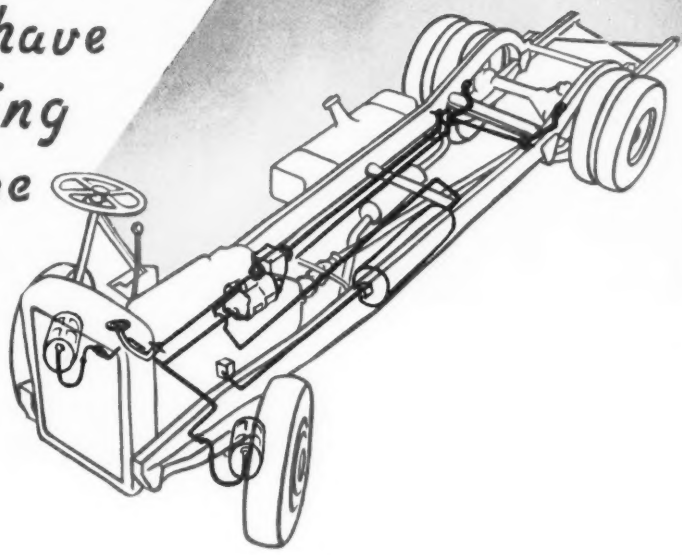

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Simplicity of layout makes installation costs small.
Method of construction and manufacture ensures reasonable costs of units
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Ease of obtaining correct braking ratio between front and rear axles.
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These features give a smooth, sure and powerful braking system that is absolutely reliable.

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Each unit independent in application.
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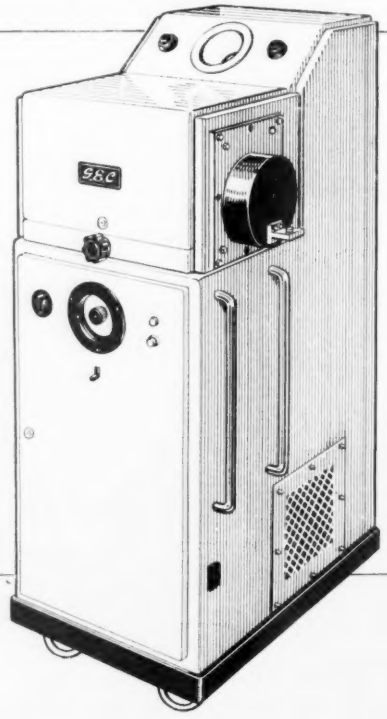
OUTPUT INCREASED 280%

When a gas flame was used to braze the box yoke the spread of heat caused such distortion that it was necessary to mill the top face and to redrill the holes. Both these operations are avoided by using H. F. induction heating, and output is increased by 280%.

The G.E.C. 2kW Induction Heater used in this application has many features which make it eminently suitable for a wide range of applications, including the hardening of small parts, soldering and brazing. Details of the generator are given in leaflet PH 1243.



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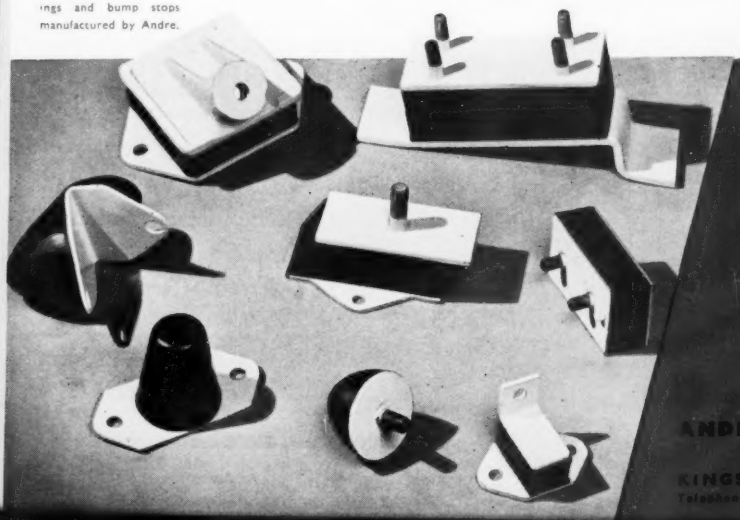
Rubber-bonded-to-metal mountings provide the most effective and economic method of damping vibration. But the design must be right and the bond beyond question. You can be sure of both at Andre where the technical service and production facilities of a company specialising in bonding and moulding rubber are at your service. Our Technical Department welcomes the opportunity to co-operate in the development of new applications of engineering rubber.

Write for a copy of "Elastomeric Engineering Examples"—a photographic record of the wide range of rubber and rubber/metal components produced by ANDRE.

A selection of Rubber-bonded-to-metal mountings and bump stops manufactured by Andre.



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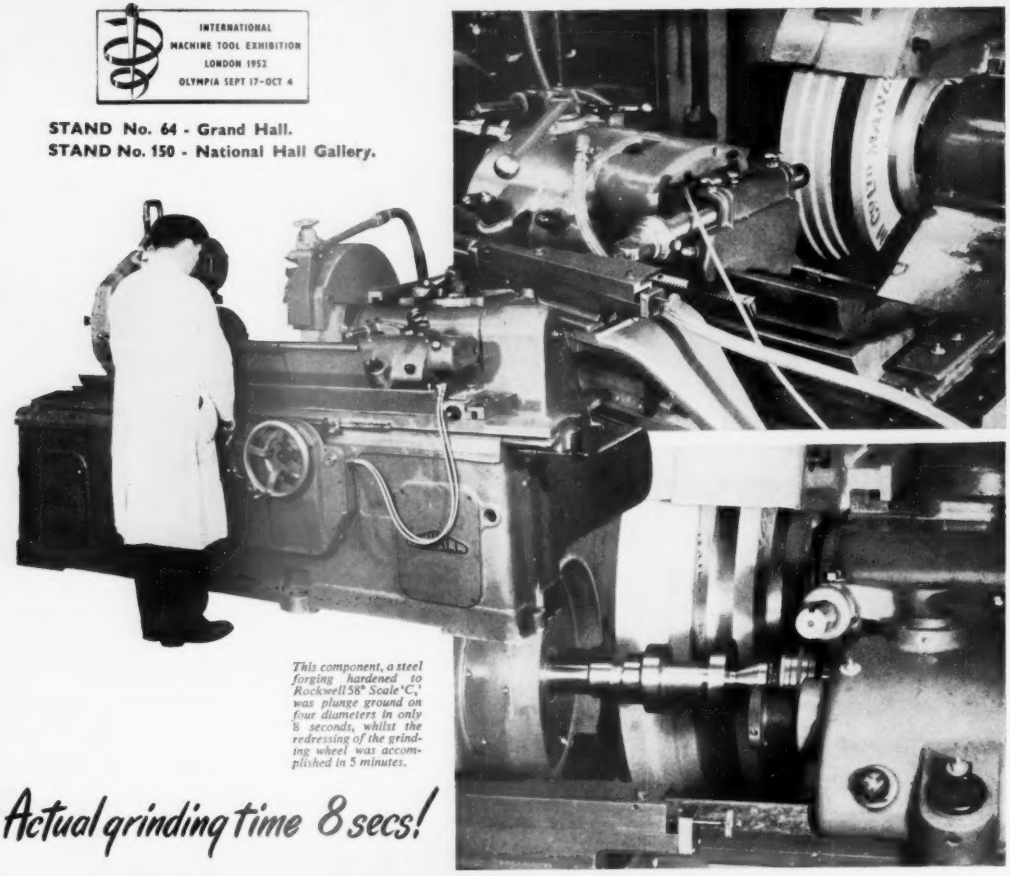
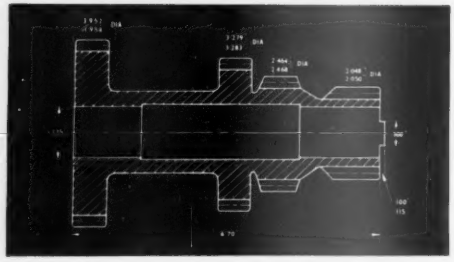


This component, a steel forging hardened to Rockwell 58° Scale 'C,' was plunge ground on four diameters in only 8 seconds, whilst the redressing of the grinding wheel was accomplished in 5 minutes.

Actual grinding time 8 secs!

Grind all diameters at one plunge..

In common with other leading automobile manufacturers, the Ford Motor Co., (Canada) Limited employ only the most modern production methods. Amongst their tools is this Newall 10″×24″ L.A. Grinding Machine. With this machine, all diameters of Gear Transmission Countershafts can be ground at one plunge—the actual grinding time taken being only 8 seconds. The L.A. Grinding Machine has proved to be the right tool for high and economic outputs.



gear transmission countershaft

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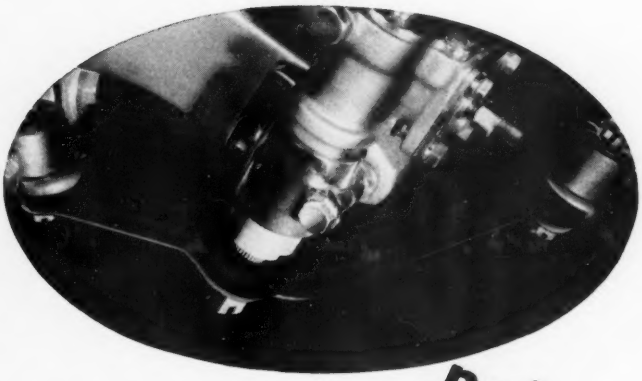
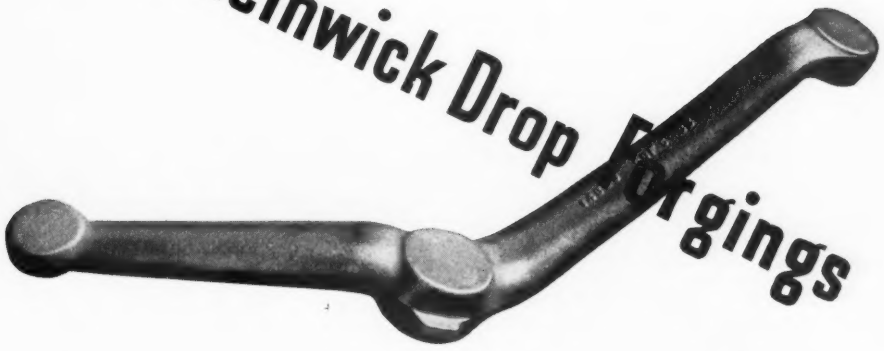
P1295



points of stress
are strong points . . .
with Smethwick Drop Forgings



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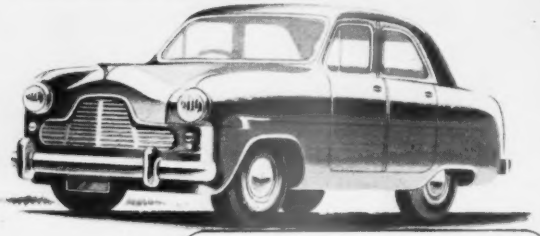
Torsion Bars & Heavy Coils



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h.r.c. fuse links

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The Association of Short Circuit Testing Authorities (ASTA) issues Certificates of Rating for fuses giving satisfactory performance when tested in accordance with the appropriate clauses of British Standard 88.

The complete range of 'ENGLISH ELECTRIC' Type 'T' High Rupturing Capacity Non-Deteriorating Cartridge Fuse Links has been ASTA certified for Category of Duty 440AC4 to the 1939 edition of B.S.88 (the highest category at that time) and for Categories of Duty 440AC4 and 440AC5 representing 25 MVA and 35MVA respectively to the 1947 edition of B.S.88.

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		440 AC 4	440 AC 4	440 AC 5
30	T1A30	617	1016	2074
60	T1S60	616	903	2022
100	TC100	615	932	2021
200	TF200	251	933	2033
300	TKF300	252	934	2032
400	TM400	629	935	2031
500	TT500	354	936	2025
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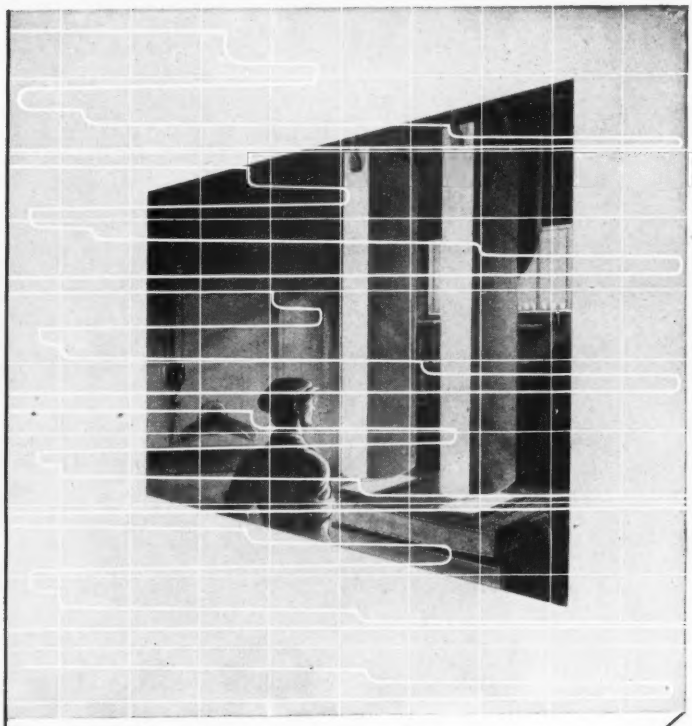

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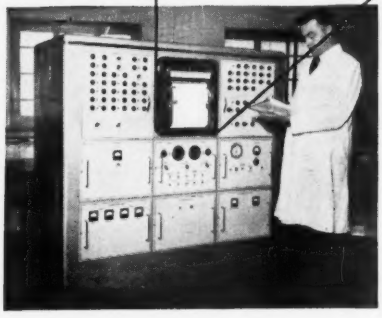

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For a quarter of a century the makers of Tru-Wel have been working towards the same goal in the manufacture of steel tubes. Although they don't pretend to know all there is to know about tubes, many of their users are kind enough to tell them they do know a good deal more than most.

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Supplied in straight lengths from ¼" o/d to 4½" o/d in gauges 22 to 6 or manipulated to your special requirements.

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Positive linkage allows a more positive gear change and the M.M.D. Spring Loaded Clevis Joint is the perfect solution to the problem of "sloppy" remote control gear change assemblies. Wear, slack and rattle on *all* mechanically operated linkages, can be eliminated throughout the life of the car, with the aid of this rationalized clevis design. Available to suit all requirements of the Motor Industry.

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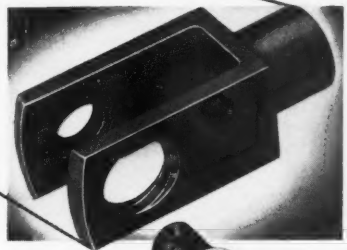

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MIDLAND MECHANICAL DEVELOPMENTS LTD.

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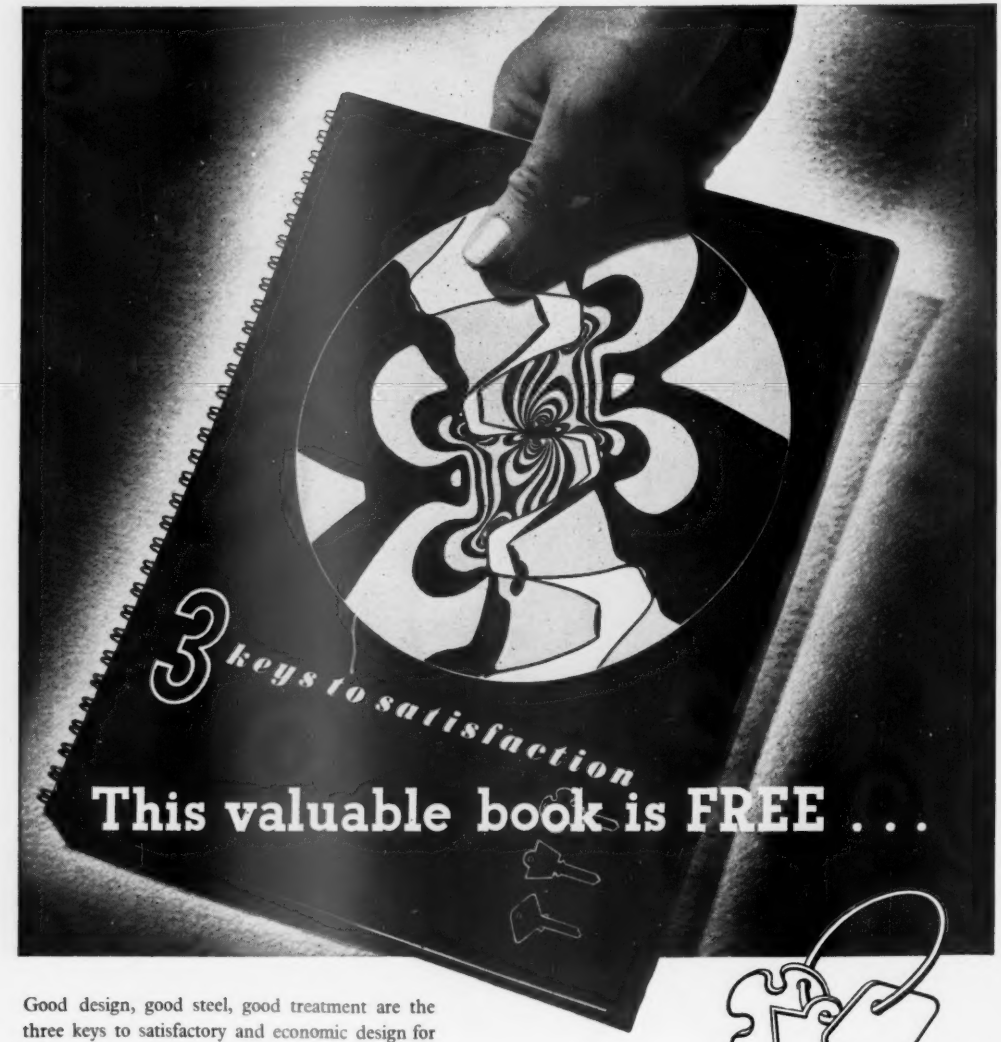
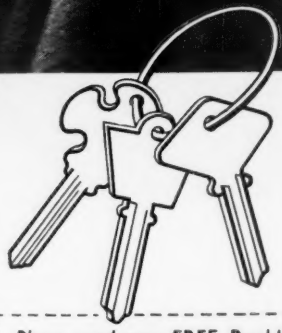


Good design, good steel, good treatment are the three keys to satisfactory and economic design for production and service. If the design is poor, then the best steel and treatment will not redeem it, and vice versa.

Every engineer, designer, draughtsman, student will welcome this valuable summary in 72 pages of the facts relating to operating stresses, stress concentrations, what alloy steel to use for different applications, its treatment, etc.

Write now—a hundred valuable diagrams, graphs and tables.



Climax Molybdenum Company of Europe Ltd.

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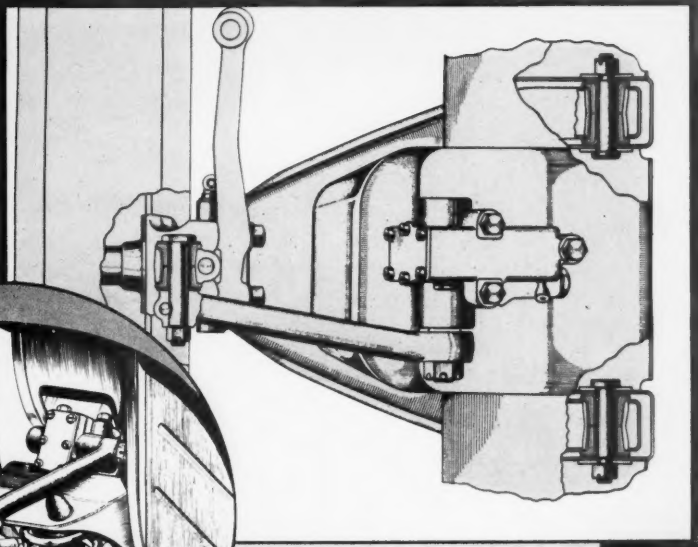

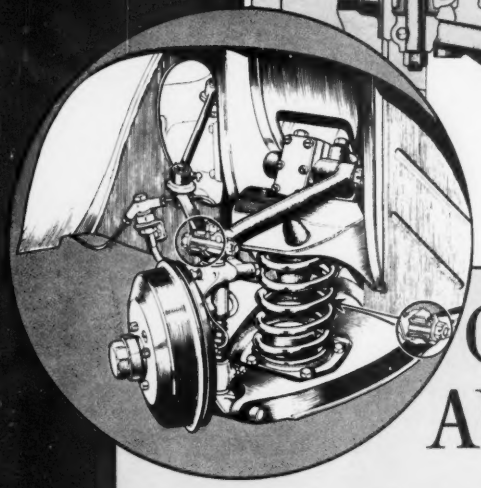

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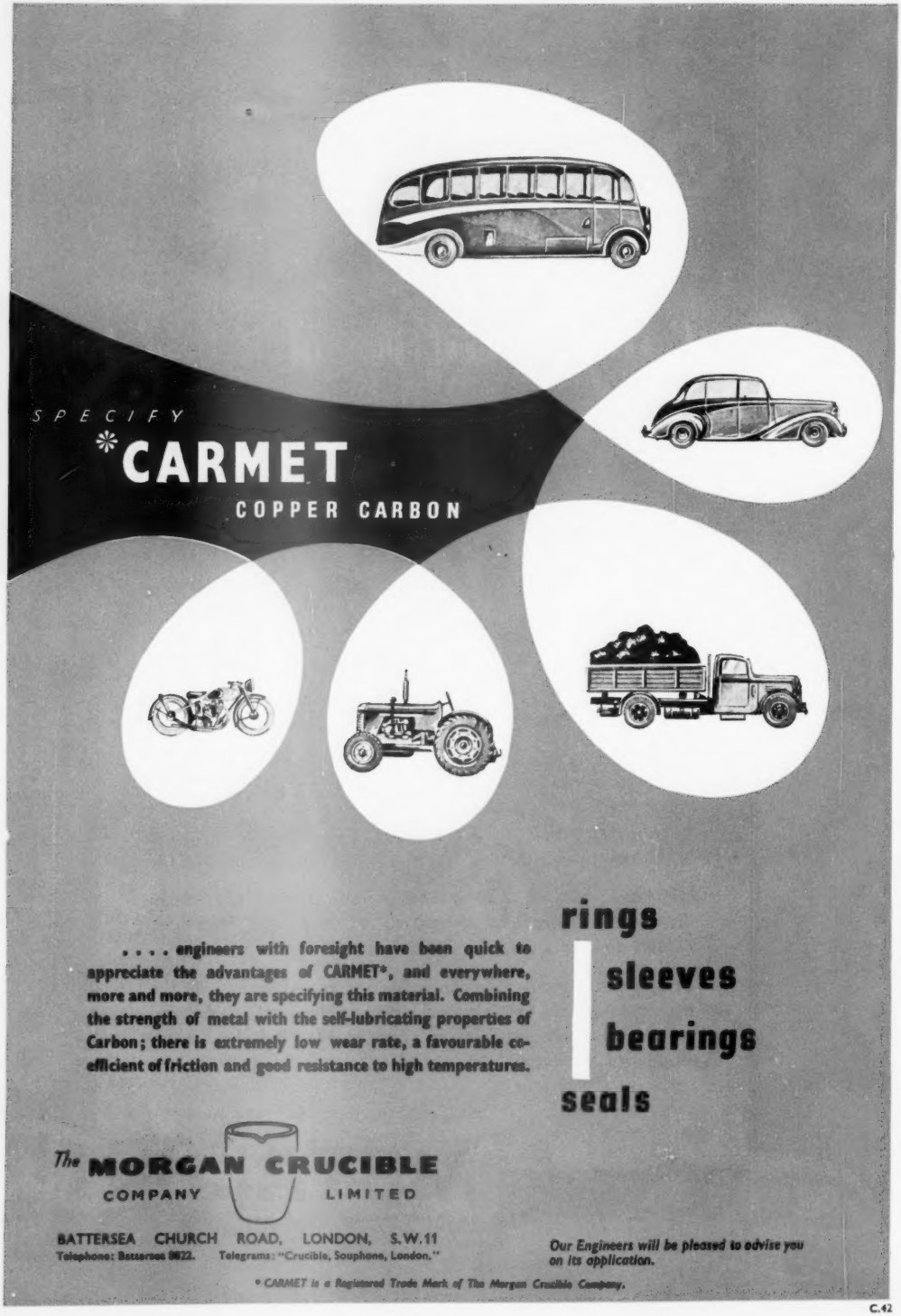
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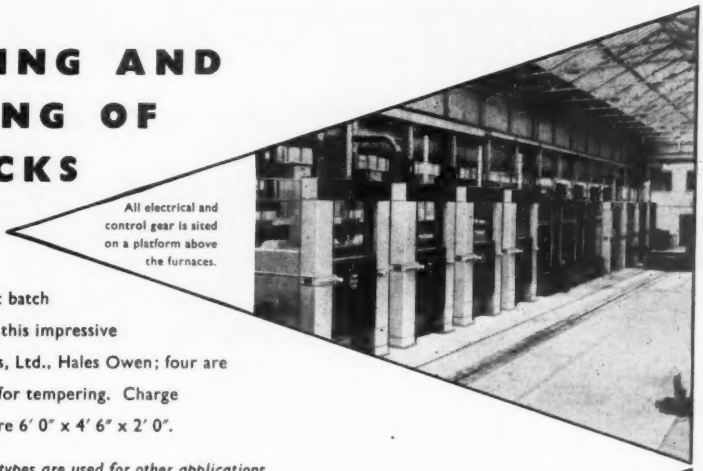

All electrical and control gear is sited on a platform above the furnaces.

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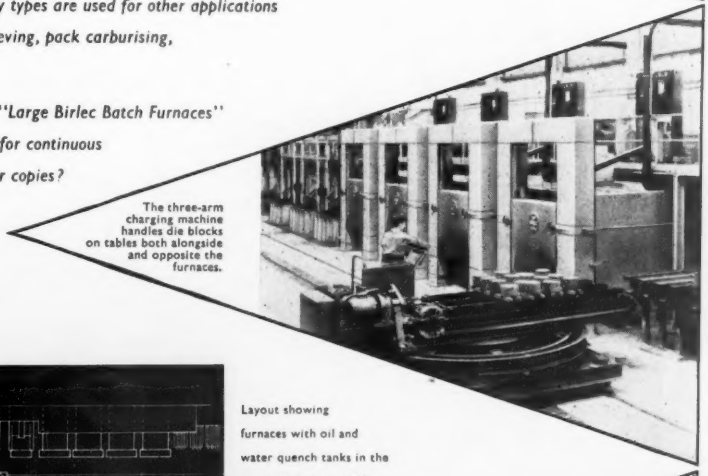
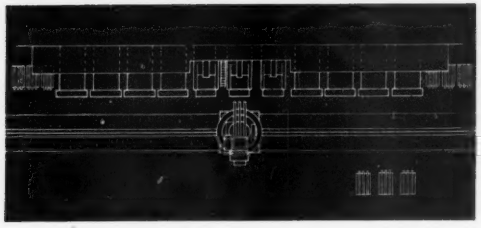
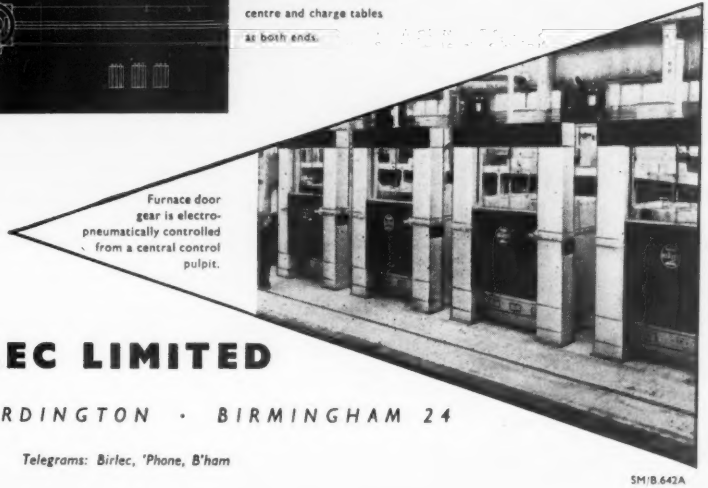


The three-arm charging machine handles die blocks on tables both alongside and opposite the furnaces.



Layout showing furnaces with oil and water quench tanks in the centre and charge tables at both ends.



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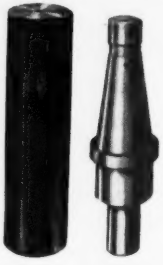

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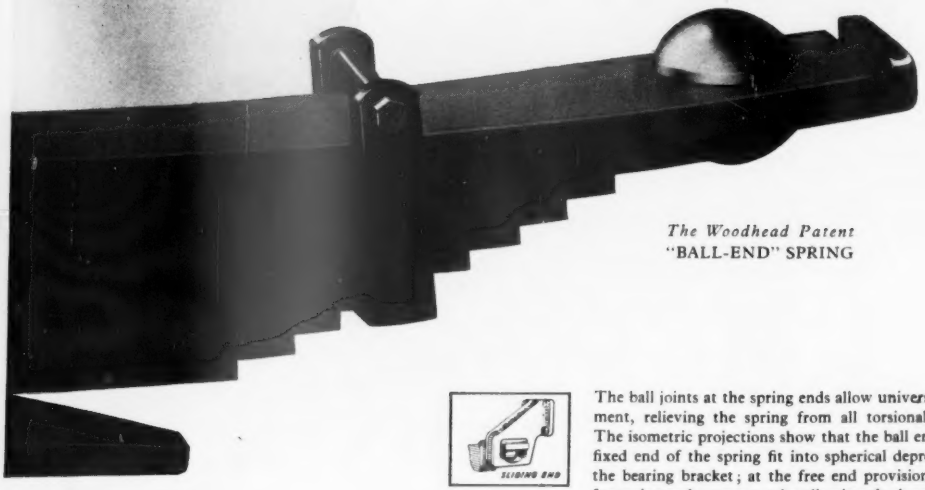

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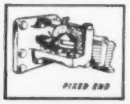

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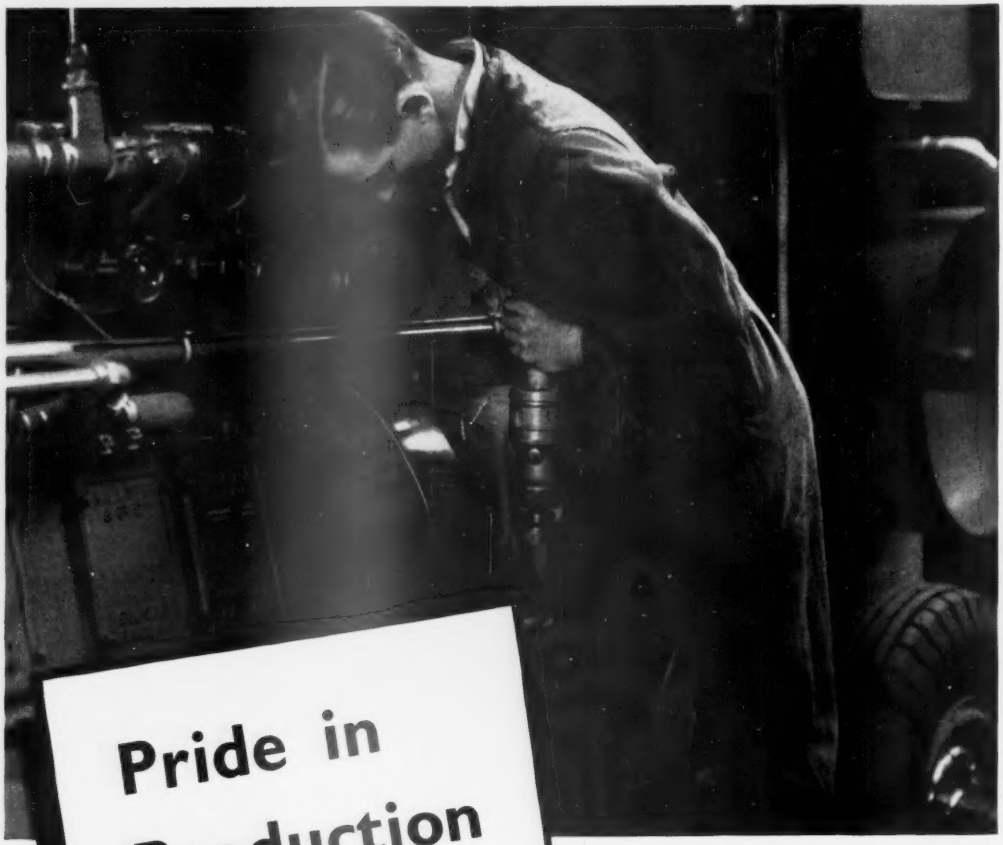

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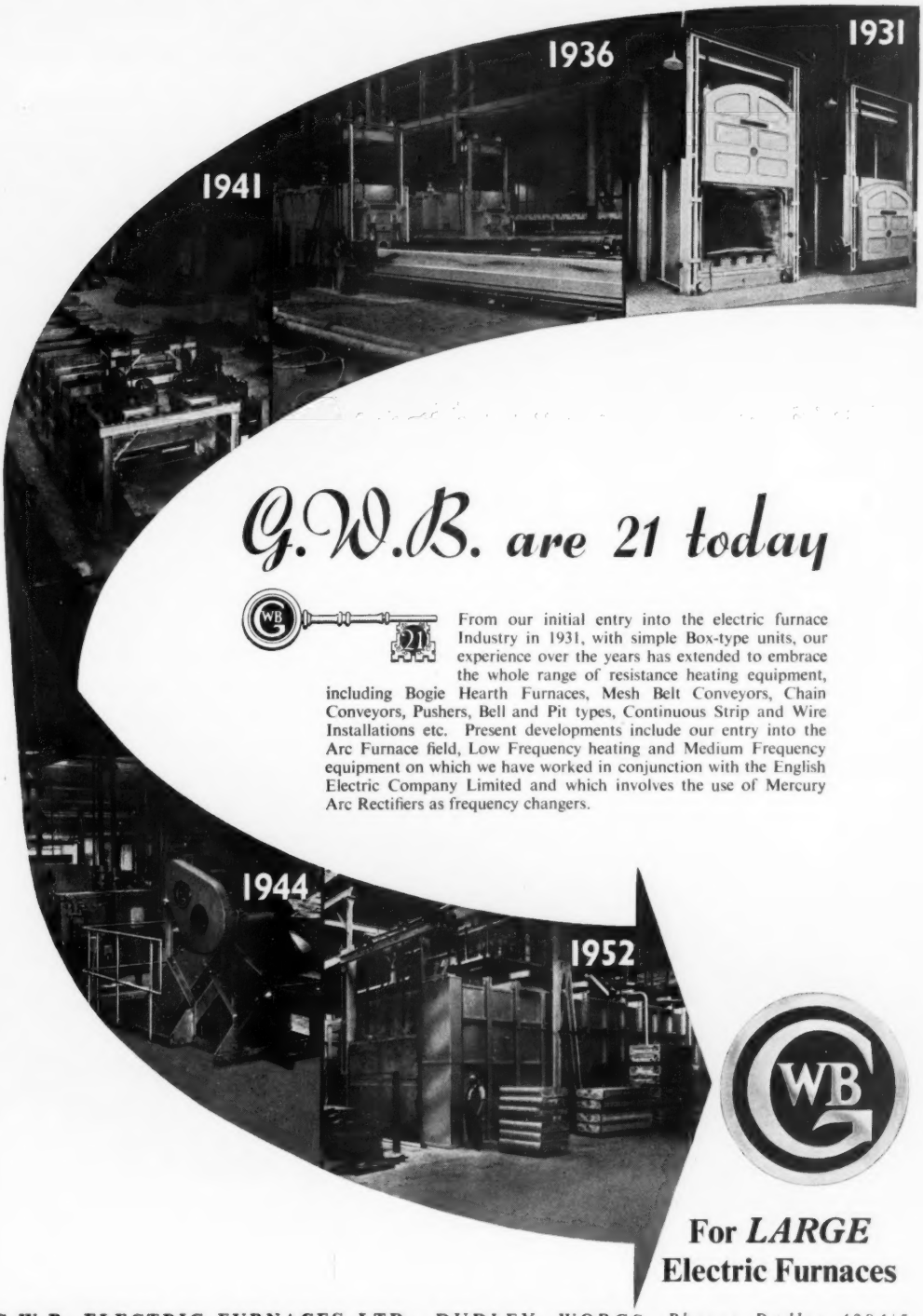

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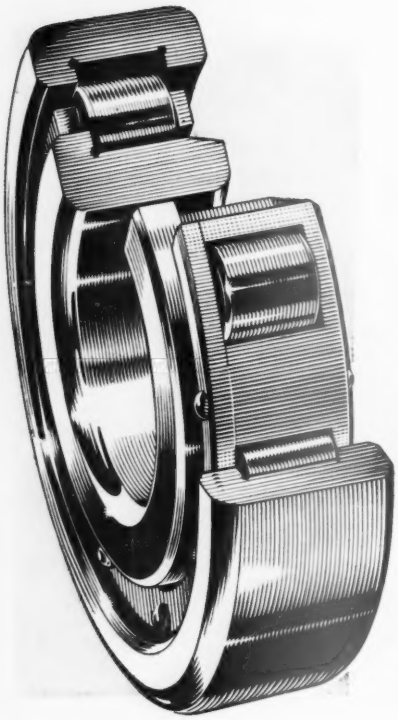

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Power Unit Developments

FROM time to time throughout the history of the piston-type internal combustion engine it has been suggested that the peak of development has been reached. For obvious reasons this may be true of piston-type engines for aircraft, which have been developed to give very high overall thermal efficiencies, but only at a cost that would be prohibitive for the automobile industry. In a lesser degree it may also be true of a few car engines. But here again, the efficiency is associated with a cost of production that must severely limit the market. Of what may be termed normal engines, it is certainly not true to say that the peak of development has been reached.

Further development is a matter of urgency. There is every indication that competition, both at home and abroad, will be very much keener in the near future. Therefore the quality and efficiency of the product will be increasingly important. Improvements in these respects can come only through continued research and development, inevitably a lengthy matter. Even when results have been achieved, there still remains their application before any tangible benefit is gained. Because of the high stage of efficiency to which the piston-type internal combustion engine has already been developed, the returns from research and development will appear meagre for the time and money spent. Nevertheless, the search for improvement must continue.

All the major manufacturers of automobiles maintain large research departments. Whether the best use is made of these departments may be questioned. In some organizations, research and development are devoted almost solely to the immediate problems concerned with current designs. This is understandable, but the importance, nay the necessity, of long-term investigations must not be overlooked. Admittedly, more research applicable to future designs in general would impose an additional financial burden, without any immediate return. This burden may appear to be unwarranted at a time such as the present, when the immediate task is to reduce costs in order to meet competition, but the additional outlay would represent only a negligible percentage of the total budget. Its effect on cost per unit should be trifling.

There is still a very wide field for investigation into means for improving the efficiency of power units. Some possible lines of approach are:—pressurized cooling systems; alternative materials and methods of construction, such as cast iron connecting rods; better lubrication and bearing materials to reduce bearing dimensions and mechanical losses; and surface discharge plugs. In sum, these might produce appreciable improvements in engine efficiency, but none, by itself, could have any very marked effect. More promising, are investigations directed towards obtaining greater power and torque from a given engine capacity. Any such increases would be the more acceptable if accompanied by a reduction in fuel consumption. In other words, the aim should be improvement in overall thermal efficiency.

A fundamental, and in many ways the simplest, means to this end is to increase the compression ratio. Unfortunately, this course is dependent upon the availability of appropriate fuels, and there is little prospect that fuels of very high octane ratings will be produced for general use for a long time to come. Manufacturers in this country, therefore, have little or no encouragement to investigate the possibility of improving engine efficiency by the use of compression ratios much higher than those now used.

Chrysler investigations

A research programme carried out by the Chrysler Corporation seems to indicate that another line of approach may yield quicker and better results. Details were recounted in a recent S.A.E. paper *New Horizons in Engine Development*, by James C. Zeder. Briefly, the Chrysler Corporation, after research into the possibilities and potentialities of appreciably higher compression ratios, turned attention to improving combustion chamber efficiency and engine breathing generally. Admittedly the results were obtained from an overhead-valve engine with spherical-segment combustion chambers, and were compared with results from an earlier side-valve engine; nevertheless when all allowances are made, the improvements in efficiency were remarkable.

Without a doubt, the spherical-segment combustion chamber approaches the ideal so far as thermal efficiency is concerned, because the surface area volume ratio is a minimum. Therefore, the heat losses to the coolant are reduced, with a consequent reduction in the size of radiator needed. Moreover, there proved to be less tendency to form a carbon deposit. But, probably the greatest advantage to be gained from this type of combustion chamber is that the lateral disposition of the valves permits the incorporation of large, unobstructed and cool ports, which give exceptionally high volumetric efficiency. Unfortunately, this arrangement of valves is scarcely

practicable for any other than the more costly in-line engines, since there appears to be no inexpensive way to operate the valves.

A glance through the engine descriptions published in past issues of this journal will show that many different forms of combustion chamber have been, and still are, used. Presumably each was chosen because in the light of all the relevant circumstances it was considered the most efficient; they cannot all be the best. Many were developed a relatively long time ago, and no doubt gave excellent results in the engines then current. In the interim, much progress has been made and conditions have changed. Porting and manifold arrangements have altered; there have been improvements in cam and valve design; the fuel position has changed (for the worse); and, as a result of recent legislation, some manufacturers have made radical changes in bore stroke ratio. All these factors can affect combustion chamber design and for every design it should be asked whether the combustion chamber has been modified to a form best suited to the existing conditions.

Engine breathing

Increases in compression ratio almost invariably entail higher friction losses; improvements in engine breathing reduce pumping losses as well as promoting high volumetric efficiency. There is therefore, good reason for a careful study of all the factors relating to the design of the induction system. Consideration must be given to cam design, valve size and shape, as well as valve guide shape and projection into the port, shape and size of the port, and the arrangement of both induction and exhaust manifolds.

Many papers on the design of cams and valve operating mechanisms have been presented before both the Automobile Division of the Institution of Mechanical Engineers and the Society of Automotive Engineers. They are worthy of careful study, since they embody a fund of knowledge. A word of warning is perhaps, necessary. Some designers have sought to apply the principles laid down in these papers, but have made mistakes in application, thereby obtaining results that led them to conclude there was no advantage to be gained. Others have effected remarkable improvements in engine performance, generally by obtaining higher valve lifts with lower accelerations, which allowed the use of lighter operating mechanisms.

A pre-requisite if valve-head diameters are to be increased, is the maintenance of uniform sections round the seats so that distortion does not occur. This problem will be eased when improved temperature-distortion resistant alloys are developed for application to cylinder head castings. Combustion chamber design also has, of course, an important effect on the size of valve head that can be employed.

In a vertical in-line engine, exhaust port and manifold design is made difficult by the fact that for maximum efficiency it would be necessary to reconcile two opposed fundamental requirements. The first is that the layout of this type of unit requires that, for greatest efficiency, the ports should be inclined upwards from the valve seats, while the second is that the only really practical path for the exhaust pipe to follow is under the vehicle.

Other engines, such as flat and vee-type, generally have the advantage so far as exhaust system lay-out is concerned. However, these types are almost invariably more expensive to produce, and may present difficulty with regard to installation. Nevertheless, a time may well come when the incorporation of some more costly features in engine design will be necessary to meet the public demand and to counter competition from abroad. After all, it cannot be denied that in recent years many other relatively expensive features have come to be regarded as essential parts of the motor car.

Machine Tool Exhibition

FROM September 17th until October 4th, 1952, Olympia, London, will be the scene of the International Machine Tool Exhibition organized by the Machine Tool Trades Association. To production engineers, this quadriennial event is rivalled in importance only by the biennial Mechanical Handling Exhibition. This year, and for the first time since 1934, the exhibition will be truly international in character, since once again the products of ex-enemy countries will be shown, in addition to machines from all other major machine tool manufacturing countries.

It will be interesting to compare British machine tools with their German counterparts. Since the war, the British machine tool industry, under difficult conditions arising from shortages of materials and labour, has done remarkably good work in equipping British engineering factories and at the same time maintaining a healthy export trade. The full order books have probably retarded developments in some measure, but nevertheless advance information concerning the products that will be displayed shows that the British machine tool industry is fully aware of its responsibilities. A pre-view of the exhibition will be published in the September issue of this journal.

CONTENTS

Page

Editorials 283-284
Power Unit Developments. Machine Tool Exhibition

The Allard P2 Chassis 285-292
A New Estate Car Chassis Designed for Cross-country Work

Engine Performance. By J. G. Lavender and C. R. Webb 293-299
An Analysis in Terms of Inlet Manifold Conditions

A Power Brake Unit 300-302
The Bendix Treadle-Vac for the Packard Easamatic System

Engine Manufacture 303-309
Recent Production Developments at the Works of Vauxhall Motors Ltd.

Page

Facel-Métallon Bodywork. By A. G. Douglas Clease 310-315
French Methods for Coachwork in Light Alloy, Steel, or Combinations of the Two Materials

Recent Publications 315-316
Brief Reviews of Current Technical Books

Noise Measuring Instruments. By W. Bye ... 317-320
Some Installations and Methods

Current Patents 321-322
A Comprehensive Review of Recent Automobile Specifications

THE ALLARD P2 CHASSIS

A New Estate Car Chassis Designed for Cross-country Work

TO meet current demands from overseas, the Allard Motor Co. Ltd. are producing an estate car incorporating many notable features. The demands have come not only from individual motorists, but also from firms hiring out vehicles for safari work in overseas territories, particularly Africa and South America. By current standards this potential market is not, of course, a large one, but it is ideal for manufacturers of specialist vehicles. While mass-produced cars may be used over terrain such as jungle tracks and unmade mountain roads, they are far from ideal for this work.

This model is outstanding because of its adaptability to many different conditions. For use on muddy or sandy terrain, large section rear tyres, 7.50 × 16.00 in may be fitted in place of the standard ones, 6.25 × 16.00 in. Moreover, alternative shock absorbers are available. They are the 1½ in bore, and the 1⅜ in bore Armstrong telescopic units. Any one of four different axle ratios can be incorporated, so that the vehicle may be supplied to suit customers' requirements and local conditions.

An essential on this type of vehicle is a sturdy frame, and there is little doubt that this requirement is satisfied by the Allard P2 frame, the torsional stiffness of which is, the manufacturers claim, five times that of the P1. In spite of its unusual strength and stiffness, the dry weight of the vehicle is stated to be only 29¼ cwt, and the curb weight is about 30 cwt. The ground clearance is 7½ in fully laden, while, with full tank, a driver and one passenger, the body floor at the front is 15 in above the ground. This is 3½ in lower than the P1. To give more room in the body, the engine has been moved 9 in further forward.

SPECIFICATION

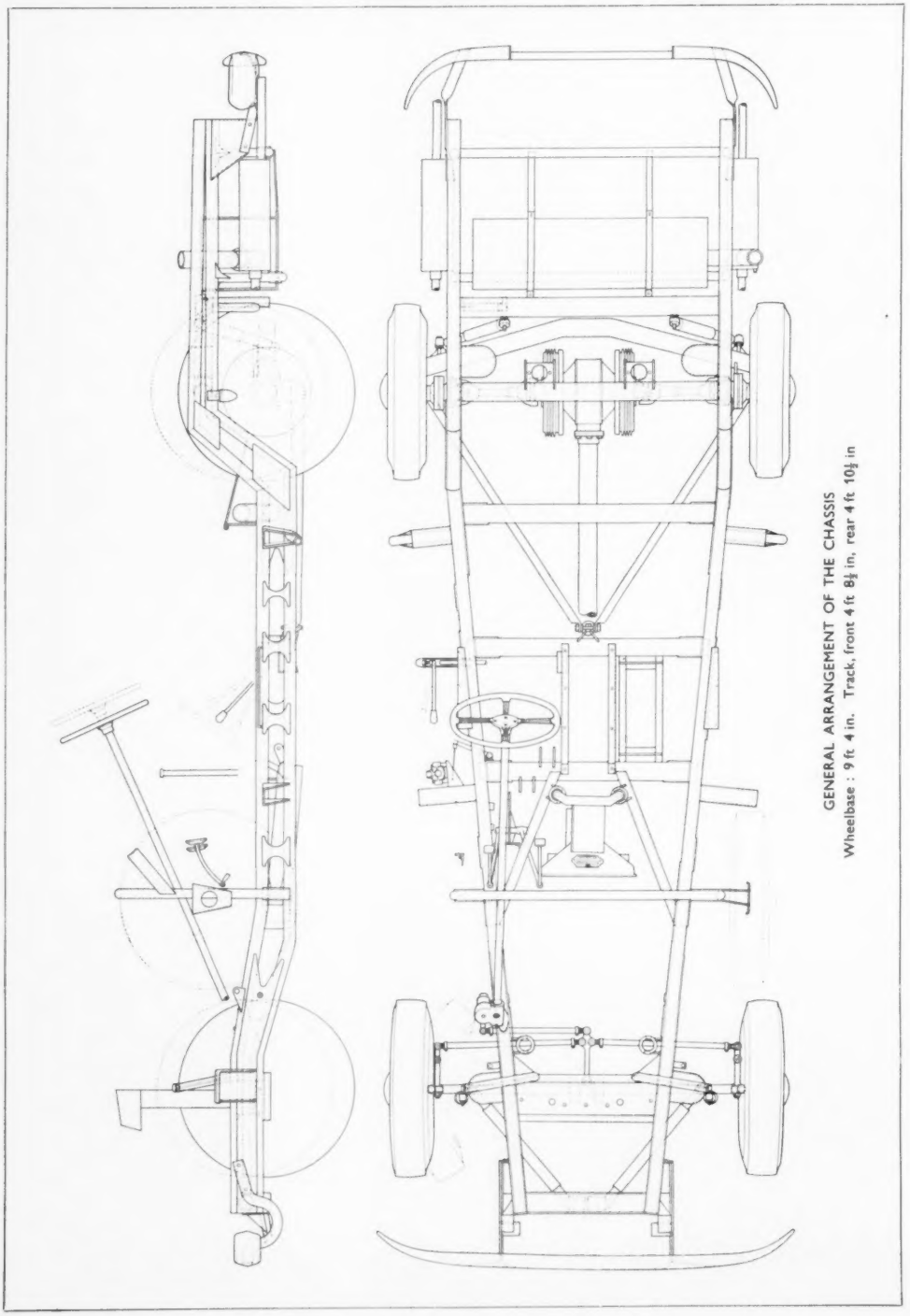
TRANSMISSION: With Ford V8 Pilot engine the gear ratios are: top 1:1, second 1·77:1, first 3·1:1, reverse 4·0:1. Ford torque tube enclosing a Hardy Spicer propeller shaft.
REAR AXLE: De Dion with swinging half-shafts. Spiral bevel final drive. Ratios available: 4·1:1, 3·78:1, 3·5:1, or 3·27:1.
FRONT SUSPENSION: Single transverse link I.F.S. with forward pivoted radius rod and coil spring.
REAR SUSPENSION: Coil spring on De Dion tube, forward pivoted radius rods, and a Panhard rod.
SHOCK ABSORBERS: Armstrong AT7 telescopic front and rear.
STEERING: Marles cam-and-roller unit, 3½ turns from lock to lock.
BRAKES: Lockheed hydraulic, 12 in diameter × 1¾ in width.
TYRES: 6.25 × 16.00 or, at the rear only, 7.50 × 16.00.
DIMENSIONS: Wheelbase 9 ft 4 in Track, front 4 ft 8½ in, rear 4 ft 10¼ in.

Unusually free access to the power unit is obtained, by hinging the entire bonnet and wing unit at the front, the pivot being at bumper level. On this model, however, two hydraulic jacks, mounted on brackets, one on each side frame are incorporated to lift the bonnet. They are powered by a Smith's Jackall hand pump bolted to the body mounting bracket adjacent to the driver's seat. A knurled thumb-screw control is incorporated in the pump to lock the hydraulic fluid in the system and hold the bonnet up, or to release it again at will. The bonnet is held in the down position by a spring-loaded, hook-type catch which may be released by a pull control under the dash.

The standard power unit is the 85 b.h.p. British Ford V8 Pilot engine, as described in the October, 1945 issue of *Automobile Engineer*. This unit gives a power : weight ratio, based on kerb weight, of 59·4 b.h.p./ton which, with a suitable axle ratio, should result in fairly good all round performance for this class of vehicle. The gear ratios are : top 1:1; second 1·77:1; first 3·1:1; and reverse 4·0:1. As an alternative, any one of several engine and gearbox units of American manufacture may be fitted for export. They include the Ford V8, the new Lincoln overhead-valve V8, the Chrysler Firepower, described in the March, 1952 issue of *Automobile Engineer*, the Cadillac 331, and the Ardun.

One of the prime requisites for selling cars overseas is the establishment of a good spare parts service.

The estate car, a difficult subject for the stylist, has pleasing lines



GENERAL ARRANGEMENT OF THE CHASSIS
Wheelbase : 9 ft 4 in. Track, front 4 ft 8½ in, rear 4 ft 10½ in

This is difficult when only a small number of vehicles is exported to each country. Allards have overcome this problem very satisfactorily by installing power units and gearboxes of mass-produced vehicles, and by fitting Ford V8 Pilot transmission and other components. It has, of course, been necessary to modify these in some respects to suit this particular installation, but parts, such as oil seals, that normally require replacing during the life of the vehicle, are standard.

Rear axle

The drive is transmitted to the De Dion-type final drive unit by a Hardy Spicer propeller shaft, enclosed in a Ford torque tube which is shortened by cutting and welding. Not only does this give the self-balanced system necessary, with this kind of axle unit, to react engine torque but, together with the frame cross member over the rear axle, it also takes the brake torque reaction.

A Ford E62A spiral-bevel differential unit is employed. It is modified to suit this layout by considerably reducing the length of the half shafts, and by manufacturing different side covers to house the differential bearings as well as to carry the brake back plate. The differential casing and the side covers are made of D.T.D. 304. In addition to the two final drive ratios available with the Ford unit, gears made by David Brown and Sons, Ltd., may be fitted to give two more alternative ratios, making a total of four in all. They are 4·1 : 1, 3·78:1, 3·5:1 or 3·27:1.

Integral with the En 26 bevel pinion is a $1\frac{1}{4}$ in diameter shaft, onto which is splined and pinned the tubular driving shaft. The pinion is straddle mounted between a two-row taper roller bearing at the front, and a plain roller bearing at the rear. Only an outer race is employed at the rear where the caged rollers run on the $1\frac{1}{2}$ in diameter portion of the shaft.

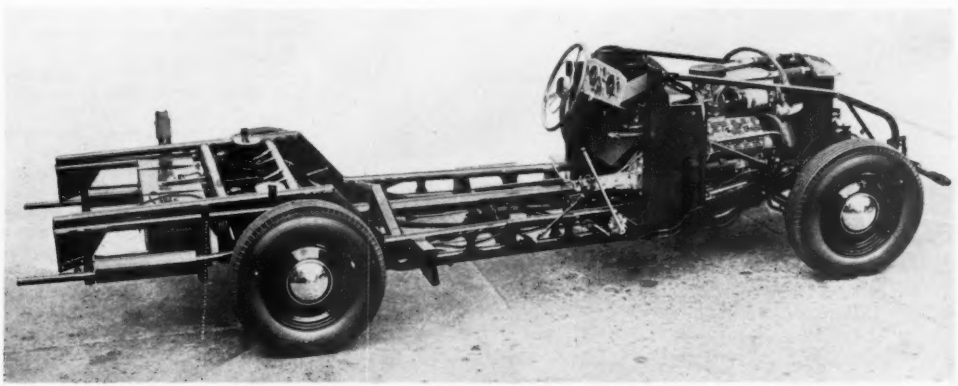
Pre-load is applied by a nut threaded on the shaft and pulling the inner races against the pinion. Another nut is tightened against the first to lock it, and, as an additional safety measure, a tab washer is interposed between the two. When the correct pre-load has been applied, the torque required to turn the pinion is 12-17 lb-in. There is no axial adjustment to the position of the pinion, but the mesh of the crown wheel is altered by fitting paper washers between the axle casing and the left-hand cover.

An En 38, $8\frac{7}{8}$ in diameter crown wheel, with teeth $1\frac{1}{4}$ in long, is employed. It is carried on a taper roller bearing around its $1\frac{11}{16}$ in diameter boss. Additional support is furnished by a similar bearing around the boss on the other side of the differential pinion cage. The outer races are carried in the side covers. Pre-load of these bearings is adjusted by fitting paper washers between the differential casing and the right-hand cover. Both covers are secured to the differential casing by ten $\frac{5}{16}$ in diameter set bolts, locked by spring washers. Some of these bolts also carry a fabricated, welded bracket for the two Metacone mountings, supplied by Metalastik Ltd. Each of these mountings is carried on a horizontal plate, $\frac{1}{8}$ in thick, welded between two vertical flanged brackets on the frame cross tube.

Six $\frac{3}{8}$ in diameter bolts with split-pinned, slotted nuts secure the two-piece cage to the right-hand side of the crown wheel. The bolts have cheese heads with flats on them registering against a shoulder on the cage to prevent them turning. The cage is split in the plane which contains the axis of the bevel pinion as well as the axes of the four arms of the spider. Each arm of the En 18A spider has two flats on its periphery to distribute oil to the bearing surface of the En 35A pinion. Spherical thrust washers of phosphor bronze are interposed between the outer faces of the pinions and the cage. At the pitch circle, the length of the tooth engagement is $\frac{5}{8}$ in.

The Ford En 8 differential wheels are integral with their shafts. No thrust washers are fitted at their flat outer faces, the wheel on the left bearing directly on the crown wheel, and that on the right against the cage. The shafts are machined $1\frac{1}{4}$ in diameter by $1\frac{13}{16}$ in long to bear, on the left, in the boss of the crown wheel and, on the right, in a similar boss on the differential cage. Outboard of their bearing surfaces in the bosses, the shafts are machined $1\frac{3}{16}$ in diameter, for a length of $1\frac{7}{16}$ in, and then tapered $1\frac{1}{2}$ deg for $2\frac{11}{16}$ in. The $\frac{11}{16}$ in diameter end of the shaft is threaded for the split-pinned, castellated nut that secures on the taper the En 6 companion flange for the inner universal coupling. A key, $2\frac{7}{8}$ in long, furnishes the drive.

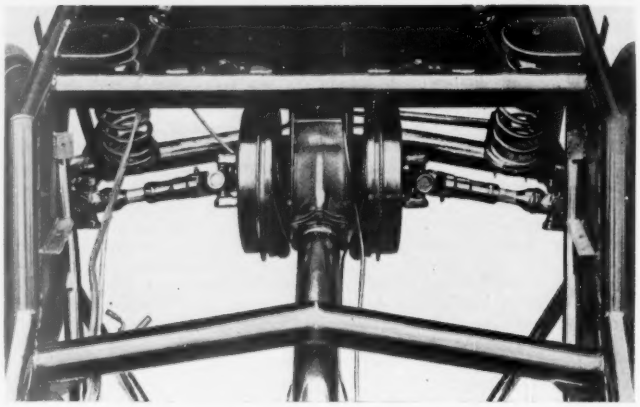
Pressed on the $2\frac{3}{32}$ in diameter inner end of the boss of this companion flange, is the inner race of a plain roller bearing. The caged rollers are located axially only by the outer race, so that the tolerance for the bedding of the boss on its tapered shaft is relatively large. A ring, secured by six $\frac{1}{4}$ in diameter countersunk set screws, clamps the outer race in its housing situated in the side covers of the differential casing. This ring also houses a Gits, lip-type oil seal that bears on a $2\frac{15}{32}$ in diameter shoulder, around the boss, adjacent to the outer face of the roller bearing. The bearing is lubricated by oil splashed through the differential gear bearing,



The engine is 9 in further forward and the floor $3\frac{1}{2}$ in lower than in the P1 model

as well as through holes, drilled in the housing to communicate between the differential casing and the space between the two bearings. Clamped between the companion flange and the universal joint is the cast iron brake drum. The assembly is secured by four $\frac{7}{16}$ in diameter studs and self-locking nuts, the universal joint being located in a spigot on the companion flange, while the brake drum is located around the same spigot.

At each end of the universal half shafts, which incorporate a splined sliding joint, a Hardy Spicer universal joint with needle roller bearings is fitted. In general, the diameter of the inner member of the sliding joint is $1\frac{1}{4}$ in, while that of the outer portion is $1\frac{3}{4}$ in. The length between the universal joint centres is about $12\frac{1}{16}$ in. At the outer end, the universal joint is spigoted into its companion flange and secured by four $\frac{7}{16}$ in diameter studs and self-locking nuts. This outer companion flange is secured in the same way as the inner one, except that it is keyed on a taper on the stub axle, and the assembly of the ball bearing around its boss is different. As well as supporting the stub axle, this bearing takes the thrust due principally to cornering loads. The inner race of the bearing is held, by a ring nut against a $2\frac{15}{32}$ in diameter shoulder on the 2 in diameter portion of the boss. The nut is locked by a radially positioned grub screw. Around the shoulder is a Gits oil seal in a housing that secures the outer race of the bearing in a recess in the En 3A wheel hub carrier, mounted on the De Dion tube.



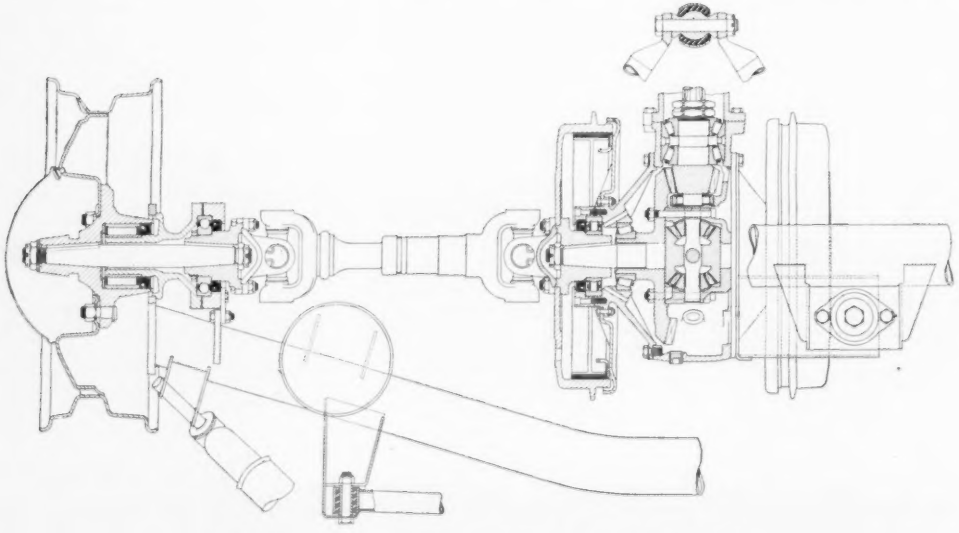
A two-point mounting is employed for the differential box

The hub carrier and stub axle are Ford units, modified to suit the Allard layout. A split-pinned, castellated nut secures the forged En 3A hub on the tapered outer end of the stub axle which is machined from the Ford half shaft. A $2\frac{5}{8}$ in long key transmits the drive. The wheel is attached, by five $\frac{9}{16}$ in diameter, special bolts and spherical-seating nuts, to a flange around the hub.

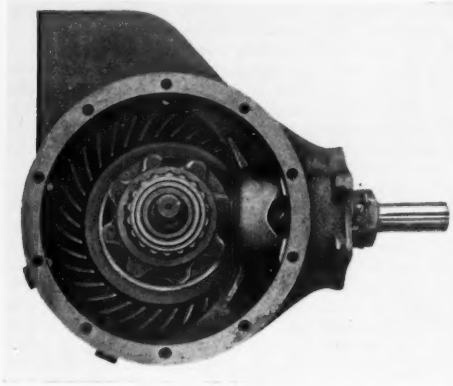
Carried in the inner end of the hub is a standard Ford bearing, the caged rollers of which are $1\frac{1}{2}$ in long. A distance washer is positioned in the housing adjacent to the outer face of the bearing, while against its inner face is a Gits oil seal located by a circlip in the housing. The inner race is mounted on the hub carrier. Lubrication of this roller bearing, as well as of the ball bearing at the other end, is effected through a grease nipple feeding through a drilling into the clearance between the stub axle and the carrier.

Rear suspension

A simple layout is employed, comprising a De Dion tube with a Panhard rod and, on each side, a radius rod. These radius rods, extending forward, are coupled together and pivoted in the vertical plane containing the longitudinal axis of the chassis. The coil springs and telescopic shock absorbers are mounted on the De Dion tube. A rubber bump stop is mounted



Most of the back axle components are either standard or modified Ford parts



The bevel pinion is straddle-mounted in its bearings

on a bracket welded to each frame side member. In the full bump position, it bears against the wheel hub carrier. A heavy canvas strap, passed around the De Dion tube, and suspended from the frame side members, forms the rebound stop.

From the axis of the rear wheels to that of the front radius rod bearings is 37½ in, while the distance from the axis of the radius rod bearings to the spring centres is 42½ in, all dimensions being taken parallel to the longitudinal axis of the chassis. This gives a spring : axle ratio of 1 : 0·885. As the actual spring rate is 260 lb in, the effective spring rate is $260 \times 0{\cdot}885^{-2}$ =330 lb in. The unsprung weight is 240 lb while the sprung weight on the rear axle is 1,560 lb. With an assumed additional load of 500 lb on the rear axle, the periodicity is approximately 105 cycles/min. Because of the 7 deg upward inclination, in side view, of a line joining the axis of the radius rod fulcrum and the axis of the wheels in the static position, there is a noticeable understeer tendency. This, of course, arises from the face that the upgoing rod moves its wheel further forward than the downgoing one on the other side.

Welded to each end of the $2\frac{3}{4} \times 10$ S.W.G. De Dion tube is a $\frac{3}{8}$ in thick plate. It is spigoted and bolted on a flange about mid-way between the two ends of the wheel hub carrier, four $\frac{1}{2}$ in diameter bolts being fitted. Spaced $2\frac{7}{8}$ in inboard of this plate is another, $\frac{1}{4}$ in thick, welded to the front of the tube. The principal function of this plate is that, together with the end plate, it takes the yawing moments that would otherwise give rise to bending fatigue on the outer plate. It is secured, by two $\frac{5}{16}$ in diameter bolts with self-locking nuts, to another plate of the same thickness. This is held in a recess in the inner face of the ball bearing housing by two of the six $\frac{5}{16}$ in diameter bolts that secure it.

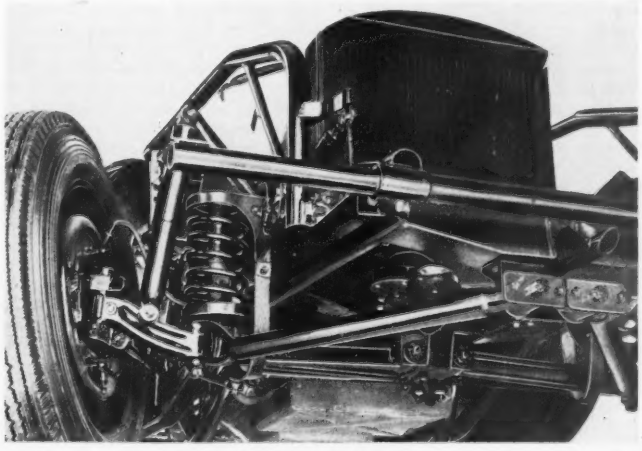
At each end, a 12 S.W.G. bracket, to support the pin for the lower end of the Armstrong AT7 telescopic shock absorber, is welded to the rear face of the tube. Rubber bushed, ring-type end fittings are employed, and they are positioned with their axes at right angles. Both are on overhung trunnion mountings, the upper pin being carried in the frame cross member.

Also welded to the rear face of the De Dion tube, but with its axis 9¼ in inward from the left-hand end, is a 12 S.W.G. bracket for the 1 in diameter × 10 S.W.G. tube which forms the Panhard rod. A $\frac{7}{16}$ in diameter fulcrum bolt, with a self-locking nut, carries the end of the rod, which is furnished with a welded-on tubular housing for a Silentbloc rubber bush. The other end of the rod is attached in a similar manner to a bracket on the frame cross member. Fabricated spring pans are supported on brackets welded to the De Dion tube. In the static position the length of the springs, as installed, is about 10½ in.

The radius rods are made from two 10 S.W.G. steel pressings welded together. At their forward ends, they form a 1¼ in diameter × 10 S.W.G. circular section to take the welded-on En 8B lugs, but their depth is increased towards the rear, to 2¼ in, the width remaining constant. Their pivoted front end assembly is held together by a $\frac{3}{8}$ in diameter bolt, the lugs being spaced approximately 2¾ in apart by the En 8 spherical centre-piece of a rubber bushed bearing. The outer portion of this, supplied by Ford Motor Co. Ltd., is of pressed steel, and is in halves bolted together on a bracket welded to a frame cross member. Welded to the rear end of each of the tubes is a plate that is secured to the hub carrier by the lower pair of the four bolts for the end plate of the De Dion tube. By virtue of their length, a little over 44 in, the greatest possible angular movement of the radius arms is only about 4½ deg down, and 7 deg up from the static position. This, together with a certain amount of flexibility of the arms in bending, ensures that the anti-roll characteristic of the layout is not excessive.

Front suspension and steering

A single-link independent front suspension unit with coil springs is employed and, whereas in previous models a rearward pivoted radius rod was employed, the drag loads are taken in this one by a forward pivoted rod. In addition, the layout differs from previous models in that the axes, about which the transverse links with their radius rods pivot, are parallel to the longitudinal axis of the vehicle. This is an improvement, as there is no inherent difficulty, with regard to variation of toe-in and castor angle, to be overcome.



The single transverse link front suspension system incorporates a forward extending radius rod to take the drag loads

The single link system was chosen for two reasons. First, it gives a high roll centre, as compared with the double wishbone link layout, and secondly, when the body and frame structure is rolling during a turn, the wheels take up, relative to the road, a more favourable attitude which, it is claimed, results in less tyre wear. This attitude is, of course, modified slightly by the sprag action, so that the outer wheel usually has only a slight positive camber, while the inner wheel is canted over in the opposite direction to the body, to give a negative camber. From the static position to full bump, the change in camber angle is about 7 deg, and the change in track, $\frac{7}{8}$ in. The normal camber and castor angles are respectively $2\frac{1}{2}$ deg and 2 deg, while the swivel pin angle is 7 deg. A toe-in of $\frac{1}{8}$ in is incorporated.

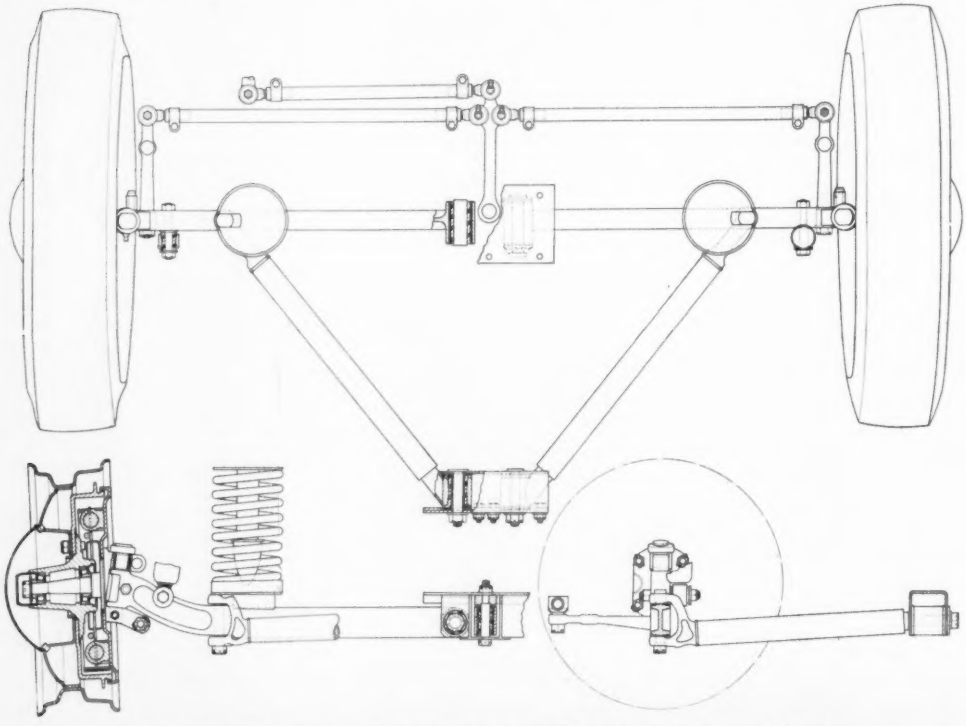
It is stated by the manufacturers that the unsprung weight of 210 lb obtained with this system is very little different from that of a double link layout for a similar vehicle. The front-end kerb weight is 1,560 lb, so that after deducting the unsprung weight, the load per wheel is 657 lb. A coil spring, having a rate of 310 lb in, is fitted with its axis 15 in from the fulcrum of each transverse link and the distance from the fulcrum to the centre of contact of the tyre with the ground is 28 in. The spring: wheel ratio is therefore 0·535:1, and the effective spring rate is $310 \times 0{\cdot}535^2 = 89$ lb in. This gives, with a static deflection of 7·6 in, a periodicity of 68 cycles min. The wheel travel is $3\frac{1}{2}$ in to bump and 3 in to rebound.

The I-section transverse links are En 19 forgings. At the inner end of each is a boss for a Silentbloc rubber bush. This is carried, by a $\frac{3}{4}$ in diameter bolt with a split-pinned, slotted nut, between two lugs of a bracket bolted under the frame cross member. Towards the outer end of the link, $16\frac{3}{8}$ in from the pivot, is a vertically drilled hole for the bolt securing the fork end of the radius rod. The bolt is $\frac{5}{8}$ in diameter, and both its round head and its split-pinned, slotted nut have a 60 deg conical face, seating in the fork end. In addition, the head has two flats on it to register in a slot in the spring pan on top of the transverse link.

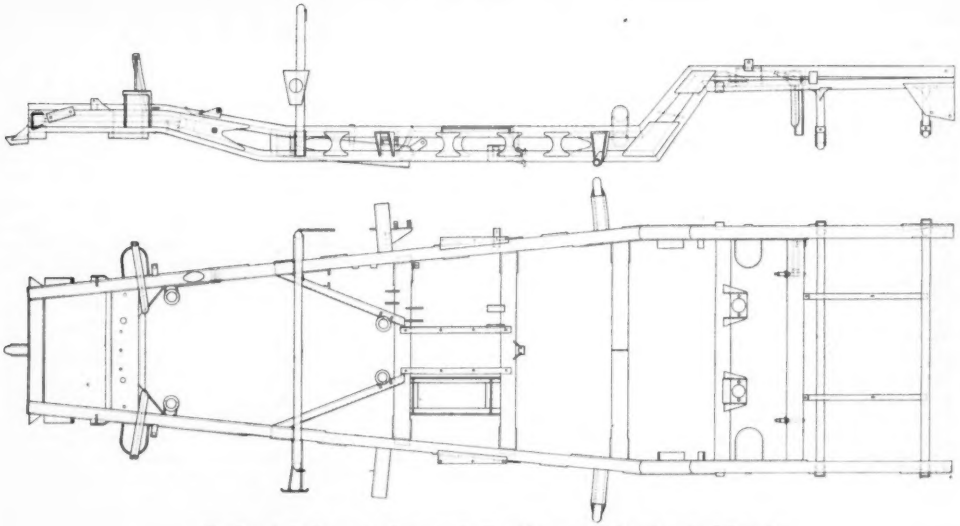
The upper end of the $4\frac{3}{4}$ in diameter spring bears on a pan-shaped bracket on the frame. A tube is welded in the pan co-axially with the spring, and it has welded to its lower end a steel disc on which is mounted a rubber bump stop. In view of the necessity for avoiding the use of pressings in small quantity production, this is probably the most economical arrangement possible. The rebound stop is a heavy canvas strap passed under the transverse link, and bolted to the upper spring pan.

A fork end on the $1\frac{3}{4}$ in diameter × 8 S.W.G. radius arm, of T40 tube, secures it to the transverse link. Welded at the other end is a boss the bore of which is tapered from each end, for a Metalastik, bonded cone, two-piece bush. The two tubes, together with the bonded-on rubber bushes, are flanged at their outer ends to take the axial thrust. A pivot bolt, $\frac{3}{4}$ in diameter is inserted from the rear, through a $\frac{3}{4}$ in diameter hole in the bracket supporting it beneath the frame front cross member. The split-pinned, slotted nut at the front end is tightened against a loose plate to compress the bush that protrudes through an inverted U-shaped, cut-out in the front lug of the bracket to which the loose plate is secured by two $\frac{5}{16}$ in diameter bolts.

Each transverse link has, at its outer end, a boss for the swivel pin, and about 3 in inboard of this is a



On each side, the front suspension links pivot about an axis parallel to the longitudinal axis of the vehicle



On the tubular frame members, nearly all welds are parallel to the axis of the tube

bolted-on trunnion mounting for the Armstrong AT7 telescopic shock absorbers. Rubber bushed, ring-type end fittings are employed, with their axes parallel, the upper end being on an overhung trunnion bolt, $\frac{1}{2}$ in diameter, passing through a tubular pedestal welded to the upper spring pan and the frame.

The swivel pin assembly is of Ford manufacture. It is located in the boss on the end of the transverse link by a taper pin secured by a $\frac{7}{16}$ in domed nut and spring washer. Between a head formed on top of the swivel pin and the upper knuckle bearing is a ball thrust bearing. Thus the wheel reaction is taken from the upper face of the knuckle to the thrust bearing and then, by the swivel pin and its locating taper pin, to the transverse link. Steel-backed, phosphor bronze bushes are pressed into the upper and lower bosses of the steering knuckle. The lower one is lubricated through a nipple, and a flat on the swivel pin carries the grease to the upper bearing. A felt seal in a dished retaining washer around the pin is interposed between the lower face of the transverse link and the lower knuckle boss to which the steering lever is secured.

Forged integrally with the En 18A steering knuckle is the stub axle. It is machined to $1\frac{3}{16}$ in diameter, and $\frac{3}{4}$ in diameter respectively for the inner and outer taper roller bearings carrying the En 3A wheel hub. Pulled up against the inner race of the outer bearing is a split-pinned, slotted nut with a large diameter washer. The thrust is taken to the outer race that bears against a flange inside the hub. From the hub, another flange transmits the thrust to the outer race of the inner bearing through which it passes to a shoulder on the stub axle. Around this shoulder is a seal, housed in the hub.

A simple, but effective steering layout is employed. Both parts of the two-piece track rod are adjustable, their end fittings, having opposite handed threads, being screwed into the ends of their tubes. They are locked by clamping rings, that are tightened by $\frac{1}{4}$ in nuts and bolts, around the tube. At the centre, they are connected to an idler lever mounted on a Silentbloc rubber bush in the bracket that houses the bearings for the front suspension transverse links. This arrangement, with the track rods very nearly in axial alignment, ensures that gyroscopic kicks or other forces transmitted from one wheel are, to a large extent, reacted by the inertia of the other wheel. The drop arm of the Marles cam-and-roller steering box is connected to the idler lever by an adjustable rod, almost parallel to the track rod. A turning circle of 40 ft is obtained with an angular movement of 28 deg on forward lock and 35 deg on rear lock. A telescopic adjustment of 3 in may be obtained on the steering wheel, the travel of which is $3\frac{1}{4}$ turns from lock to lock.

Frame

A notable feature of the frame, apart from its strength and stiffness, is that in order to avoid the use of pressed steel components it is built up almost entirely of tubular members. As a result, it is possible to manufacture all parts of the frame and assemble it at the Clapham factory; and, should modifications be required in the future, to adapt it to different vehicles, no undue restrictions are imposed by the necessity of recovering the high cost of press tools. So as to localize distortion, nearly all welded joints are arranged parallel to the axis of the tube.

The only member not of tubular section is the cross member supporting the front suspension. This is fabricated from four 10 S.W.G. mild steel plates. The vertical ones at the front and rear are both flanged at their top edges. Two more plates complete the box section, one being welded to the lower edges of the first two, and another is welded between them, $2\frac{1}{2}$ in below their out-turned flanges. Inside the box section thus formed, two additional 10 S.W.G. flat plates are welded in vertical planes, one each side of the longitudinal axis of the vehicle and parallel to it. The radiator is mounted in the channel formed by the top of the box section.

The suspension-spring pans and their supporting brackets are made of flame-cut 10 S.W.G. sheet steel and are welded to the side members, with the spring axes offset about $2\frac{1}{8}$ in to the rear of the principal axis of the cross member. Above each bracket is a $1\frac{1}{4}$ in diameter steel tube inclined with its lower end on a gusset between the front cross member and the side member, and with its upper end, carrying the front shock absorber mounting bolt, about 6 in higher than the top of the spring pan. This tube is connected by vertical welded-on

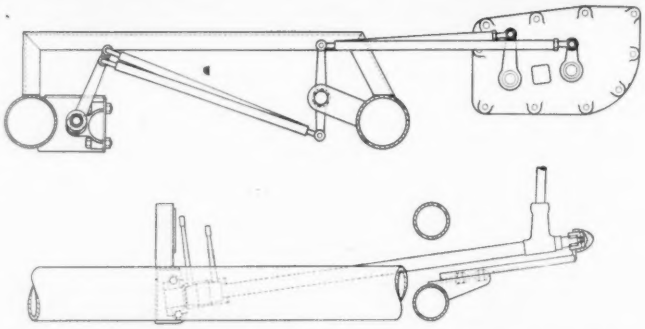
webs to the spring pan and to the gusset.

At the extreme front end of the frame, a 3 in diameter × 10 S.W.G. tube forms the cross member supporting the radius rod bearings. Five other main cross members are incorporated, two under the front seat, one immediately behind the heel board, one over the rear axle and the other, carrying the rear shock absorbers, about 13¼ in behind it. All the tubes are 3 in diameter by 10 S.W.G., but the one behind the heel board is fabricated from two pieces each registering at its outer end in a U-bracket, also in 10 S.W.G., welded to the side members. The centre, where the two inner ends of the tubes are welded together, is about 5 in above the outer ends, so providing clearance for the torque tube.

At the rear of the frame, three 1½ in diameter × 16 S.W.G. tubes are fabricated and transversely mounted, 20 in apart, to form a cradle to carry the 18-gallon petrol tank. The greater part of the weight is taken on the front one, while additional support is furnished by two longitudinally disposed members welded to the two cross tubes.

Another 1½ in diameter × 12 S.W.G. tube forms a hoop to support the dash. Mounted on one side of it is a bracket to carry the spare wheel. Beneath it, two box section frame bracing members are fitted. They are welded, at their front ends, one to each side member, and at their rear ends to the cross member, under the front of the driver's seat, where they support the rear engine mountings. The front engine mounting brackets are welded to the side frames about 5 in to the rear of the suspension cross member. Firestone rubber mountings are fitted at all four points.

The frame side members are each formed by two 2 in diameter × 12 S.W.G. tubes, positioned one above the other. They are connected by I-shaped plates, of 10 S.W.G., positioned, one on the inner side and four at irregular intervals on the outer side, between the dash and the heel board. In front of the dash a continuous plate welded on each side of the tubes forms the connection between them. Where the steering box is fitted, it has been necessary to cut away part of the upper tube. The cut-out is closed by two welded-in plates, and the section is reinforced by a plate welded to the inner face of the side member. Adjacent to the rear seat, where the tubes are each cut in two places and welded together again to carry the frame over the axle, lozenge-shaped interconnecting plates are employed at the outer face by the welded joints, and a continuous plate is welded to the inner face of each member. At the extreme rear, the tubes are joined by two more plates, one on each outer face. These plates are extended downwards and drilled to carry the bolted-on bumper irons.



A Ford steering column gear shift control is mounted transversely under the front floor. Side elevation (upper) and end elevation (lower)

Apart from the dash support hoop and eight subsidiary supports, four main body mounting points are provided. These are outrigger brackets, two on each side of the frame between the dash and the heel board. The subsidiary points are as follows. The floor is clamped between the front seat runner and a bracket on the frame side member. Four more brackets, two on each side member, support the seat pan. The back of the body is steadied by a lug adjacent to each spring pan, and by a bolted connection to the extreme rear of both side members. Welded to the rear pair of outrigger brackets are tubes for the Smith's Steadylift jack. The front end of the car may be lifted by a similar tube in a bracket welded to the centre of the front cross member.

Other features

Lockheed hydraulic brakes, with 12 in diameter drums, are employed at both the front and the rear. The shoe width in all brakes is 1¾ in, and the total brake lining area is 164 in², that in the front brakes being 83 in². A Girling, pistol-grip, hand brake control is fitted under the dash. It is connected to the end of a lever pivoted in a bracket welded to the outer end of the frame cross tube. Attached to the lever between these two connections, to give a lever ratio of 3·2:1, is a Bowden cable. At the other end of the cable its inner member is pinned to the operating lever mounted on the left-hand back plate, while the outer casing is carried in the end of the operating lever on the right-hand back plate, an involute spring being interposed, around the inner cable, between the two levers.

Beneath the brake pedal, the master cylinder is bolted to two brackets welded on the inner face of the frame side member. The two ⅜ in diameter bolts, securing the unit, also clamp on its inner side, a 10 S.W.G. flat plate to which is welded a tubular boss for the spindle carrying the pedals. On the front end of this plate, another one is welded in a transverse plane. It is bolted to two brackets, one on the side frame, and one on the bracing member carrying the rear engine mountings. The brake and clutch pedals are positioned one each side of the boss which is supported by gusset plates. Vandervell phosphor bronze bushes are pressed into the bosses of the pedals and grease nipples are fitted for lubrication. The brake pedal travel is 5 in, and the lever ratio is 4·65:1.

A modified Ford steering column gear shift mechanism, described in the September, 1951 issue of the *Automobile Engineer*, is fitted, but it is not on the column. Instead, it is mounted on the floor, outboard of the driver's seat, so that the centre of the floor is left unobstructed. The system is designed to give a positive feel to the gear change. A bracket on the frame side member supports the hand-controlled end of the mechanism, and the spindle, which in the normal installation would pass down the steering column, is shortened and positioned transversely in the frame. The inner end is carried on a bracket, welded to the frame cross member under the rear of the driver's seat. In order to reverse the direction of motion, the two rods are taken forward and attached to a pair of centrally pivoted levers mounted between two lugs welded to the frame cross member under the front of the seat. The other ends of these levers carry the rods that are connected to the selector control in the gearbox.

ENGINE PERFORMANCE

An Analysis in Terms of Inlet Manifold Conditions

By J. G. Lavender, Ph.D., B.Sc.(Eng.), A.M.I.Mech.E. and C. R. Webb, B.Sc.(Eng.), A.M.I.Mech.E.

IT is well known that conditions in the inlet manifold can have a great influence on the performance of a spark ignition engine. The investigations reported here were made on a small four-cylinder engine of contemporary design in order to determine the relationship existing between mixture flow, manifold depression, and temperature. A simple general theoretical analysis is presented and the test results for the particular engine investigated are compared with this analysis.

It will be shown that the existence of the critical pressure ratio over the carburettor throttle valve limits the power output and the no-load speed of the engine. Furthermore, when the engine is driven on the overrun, values of manifold depressions and temperatures are much higher than those attained during normal driving conditions. The fuel consumption on the overrun is shown to remain constant at any particular throttle opening regardless of speed. The assumption, frequently advanced, that the torque varies linearly with manifold depression independent of speed is shown to be inaccurate, especially at no-load and overrun conditions.

During the normal operation of an automotive engine the greater part of the running time is performed under part-throttle conditions and only in such cases as sudden acceleration or steep hill climbing is the full throttle position used, Ref. 1. It is also probable that 25 per cent of the running time, especially in heavy traffic conditions, may be performed on the overrun. Consequently, the investigations have included tests at all throttle openings with the engine on power and being driven on the overrun.

In addition to influencing the power output of the engine, the manifold depression is often used to operate such devices as the vacuumatic ignition advance and the clutch withdrawal mechanism. It is necessary from that viewpoint, therefore, to know the exact nature of the varations in manifold depression that occur under different operating conditions in order to permit accurate design of such auxiliary devices.

Significance of the manifold conditions

Manufacturers have investigated the effects of manifold design upon such factors as friction losses in the manifold, mixture distribution and the associated degree of vaporization of the fuel before entering the cylinders. Certain criteria have been laid down as to the exact requirements of an inlet manifold. Nevertheless, however well a manifold is designed there will always exist in that manifold certain conditions of pressure and temperature that will ultimately control the power output of the engine.

The power output of the petrol engine is normally controlled by a throttle valve placed downstream from the carburettor choke and this valve, in metering the weight of mixture supplied to the engine, regulates the pressure and density of the mixture of air and petrol vapour in the manifold.

In order to draw the mixture through the throttle valve a small amount of energy must be supplied by the engine, and at very small throttle openings this amount of energy may become quite appreciable in comparison with the total available power. Consequently, in reducing the power output of the engine by closing the throttle valve several factors are involved: the weight of charge entering the cylinders is reduced, the negative or pumping work is increased and the rate of evaporation of the fuel droplets is increased owing to the lower manifold pressure. This rate of evaporation of the fuel and the drop in pressure across the throttle valve will directly affect the temperature of the ingoing mixture and this, in turn, will affect the volumetric efficiency and hence the power output of the engine. The temperature of the manifold air is itself dependent upon the amount of preheating (if any), afforded to the carburettor air, the "hot-spotting" provided in the manifold, and finally the latent heat of evaporation of the fuel. Thus the conditions of pressure and temperature existing in the inlet manifold under fixed operating conditions are interdependent and it is the combination of these conditions that will finally determine the power output of the engine.

The following theoretical considerations are not intended to form a rigorous solution of the problem but are presented as an indication of the variations in the cycle of a four-stroke petrol engine, consequent upon changes in inlet manifold conditions.

SYMBOLS AND DEFINITIONS

V_s = swept volume, ft³/cylinder
V_c = clearance volume, ft³/cylinder
V_T = (V_s+V_c) = total volume, ft³/cylinder
V_o = swept volume during valve overlap, ft³/cylinder
r = compression ratio
W_s = weight of air drawn in, lb/cylinder/cycle
W_R = weight of residuals, lb/cylinder/cycle
W = weight of air consumed, lb/hr
w_f = weight of fuel used, lb/cycle/cylinder
w = weight of fuel consumed, lb/hr
P_m = manifold absolute pressure, lb/ft²
P_a = atmospheric pressure, lb/ft²
T_2 = temperature during suction stroke, deg C (absolute)
C = calorific value of fuel, C.H.U/lb
ξ = combustion efficiency or ratio of heat in fuel utilized to calorific value of fuel
K_v = specific heat at constant volume of cylinder gases
A_T = projected area of throttle valve opening, ft²
C_D = discharge coefficient over throttle valve
γ = index of adiabatic expansion for air
R = universal gas content ft-lb/lb/deg C.

Cylinder conditions

Fig. 1 indicates the theoretical indicator diagram for a 4-stroke petrol engine. In order to simplify the problem the exhaust has been assumed to take place at atmospheric pressure and it has also been assumed that all the fuel is burnt at constant volume at the end of the stroke. No account has been taken of the effects of dissociation during the combustion period and the specific heats of the gases have been taken as constant. The compression and expansion curves of the diagram have been assumed to be adiabatic with only air present in the cycle.

Corrections may be made to the final equations to allow for any of the above effects but, in general, as results will show, this approximate analysis gives quite accurate results.

For the cycle represented in Fig. 1 the negative work per cycle is given by the area 1, 2, 6, 7, 1 and the positive work by the area 2, 3, 4, 5, 2. At the end of the exhaust stroke the residuals remaining in the cylinder occupy a

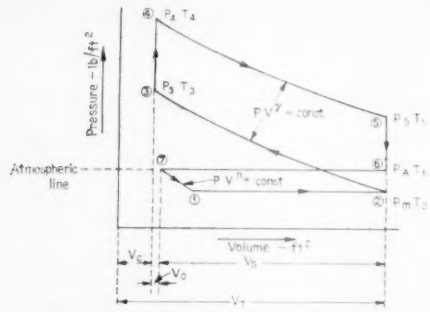


Fig. 1. Pressure-Volume diagram for one cylinder

volume V_c and will be at the exhaust temperature T_7. During the first 20 deg of crank rotation commencing the suction stroke, both the inlet and the exhaust valves are open and hence, disregarding the inertia effects of the moving columns of gases in the manifolds, some of the exhaust gases will tend to flow through the cylinder to equalize the pressure in the inlet manifold. This has been represented on the P-V diagram by the atmospheric line until the final closing of the exhaust valve. The residuals remaining in the cylinder after the exhaust valve closes will then expand according to the law PV^n = constant from the assumed atmospheric pressure to the pressure in the inlet manifold, and will occupy a volume V_1 at this pressure. Thus the effective swept volume of the cylinder will be $(V_T - V_1)$.

As the engine speed increases the ramming effects in the induction and exhaust manifolds will increase and the effect of the valve overlap will be to increase rather than decrease the effective swept volume up to a certain speed dependent on the overlap used, Ref. 2.

Neglecting any positive work done by the expansion of the residual gases, the negative work done by the engine in drawing in the fresh charge will be given by :

$$(P_A - P_m)(V_T - V_1) \text{ ft-lb/cycle/cylinder}$$

Also: Compression work done

$$= \left[\frac{P_3 V_c - P_m V_T}{\gamma - 1}\right] \text{ ft-lb/cycle/cylinder}$$

Expansion work on the piston

$$= \left[\frac{P_4 V_c - P_5 V_T}{\gamma - 1}\right] \text{ ft-lb/cycle/cylinder}$$

Exhaust work done

$$= 0$$

Work done by the engine overcoming frictional resistance = F ft-lb/cycle/cylinder

Thus the net work done

$$= \left[\frac{P_4 V_c - P_5 V_T}{\gamma - 1}\right] - \left[\frac{P_3 V_c - P_m V_T}{\gamma - 1}\right] - (P_A - P_m)(V_T - V_1) - F \text{ ft-lb/cycle/cylinder}$$

and this may be expressed as:

$$BHP = \frac{\text{No. of cylinders} \times \text{r.p.m.}}{33{,}000 \times 2}\left[w_F \xi C \left\{1 - \left(\frac{1}{r}\right)^{\gamma-1}\right\} - (P_A - P_m)\left\{V_c r - (V_c + V_o)\left(\frac{P_A}{P_m}\right)^{\frac{1}{n}}\right\} - F\right] \quad (I)$$

using the symbols as previously defined.

When the engine is on no-load the BHP = 0, therefore,

$$w_F \xi C \left\{1 - \left(\frac{1}{r}\right)^{\gamma-1}\right\} = (P_A - P_m)\left\{V_c r - (V_c + V_o)\left(\frac{P_A}{P_m}\right)^{\frac{1}{n}}\right\} + F$$

and for a four cylinder engine this may be written as,

$$w = \frac{\text{r.p.m.} \times 120}{\xi C\left\{1 - \left(\frac{1}{r}\right)^{\gamma-1}\right\}}\left[(P_A - P_m)\left\{V_c r - (V_c + V_o)\left(\frac{P_A}{P_m}\right)^{\frac{1}{n}}\right\} + F\right] \text{ petrol lb/hour} \quad (II)$$

The effective swept volume of the cylinder $(V_T - V_1)$ is equal to $\left[V_c r - (V_c + V_o)\left(\frac{P_A}{P_m}\right)^{\frac{1}{n}}\right]$ and thus at intake temperature T_2 and pressure P_m the weight of air consumed by the engine is given as

$$W = \text{r.p.m.} \times 120 \times \frac{P_m}{RT_2}\left[V_c r - (V_c + V_o)\left(\frac{P_A}{P_m}\right)^{\frac{1}{n}}\right] \text{ lb/hr} \quad (III)$$

If the throttle valve on the carburettor were completely closed (W = 0) and the engine driven on the overrun, the manifold pressure would therefore be :

$$P_m = P_A\left\{\frac{(V_c + V_o)}{V_c r}\right\}^n$$

and substituting the actual engine constants in this equation,

$$P_m = \frac{P_A}{5{\cdot}414^n} \quad \ldots\ldots (IV)$$

It should be noted that under these conditions there would be no ramming effects of the air and hence the speed would not affect the results.

Air flow over the throttle valve

In the following theoretical considerations the carburettor choke tube has been included as part of the restriction offered by the throttle valve. Thus the total pressure drop between the atmosphere and the inlet manifold has been taken as the pressure drop across the valve, Fig. 2.

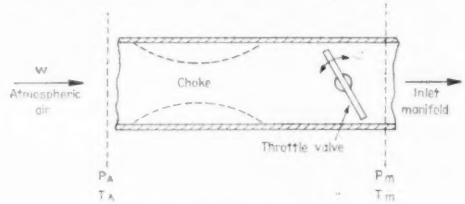


Fig. 2. Diagrammatic arrangement of throttle valve

Neglecting the fuel present in the charge and assuming frictionless, adiabatic air flow across the valve, then the weight of air flowing into the manifold for a given area of throttle opening and a given manifold depression below the critical pressure is :

$$W = 3600\, C_D A_T \sqrt{2g\left(\frac{\gamma}{\gamma-1}\right)\frac{P_A^2}{R\,T_A}\left(\frac{P_m}{P_A}\right)^{2/\gamma}\left[1 - \left(\frac{P_m}{P_A}\right)^{\frac{\gamma-1}{\gamma}}\right]} \text{ lb/hr} \quad (V)$$

When the manifold pressure is critical or below the critical value, $\frac{P_m}{P_A}$ may be replaced by $\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$ which equals 0·528 when γ is 1·404.

This results in the weight of air flowing into the manifold being given by :

$$\dot{W} = 3600\, C_D A_T \sqrt{2g \frac{P_A^2}{R T_A} \cdot \frac{\gamma . 2^{\frac{2}{\gamma-1}}}{(\gamma+1)^{\frac{\gamma+1}{\gamma-1}}}} \text{ lb hr} \ldots \text{(VI)}$$

Equation (VI) shows that if the atmospheric pressure is assumed constant, say at 30 in Hg, then the maximum air flow will occur when the manifold depression below atmospheric is 14·16 in Hg or more.

Manifold depression is caused by the rate of engine pumping. Under power conditions, a manifold pressure less than the critical would require a speed increase, but this is not compatible with the constant amount of mixture which would flow below the critical value. On the overrun, however, an external driving source can increase the rate of pumping, and pressures below the critical do occur. In fact, it has been shown that the manifold pressure can be reduced to a value $\frac{P_A}{5{\cdot}414^n}$ when the intake is completely sealed.

In the case when supercharging is used and the pressure represented in equation (V) as P_A is increased, the power output of the engine will be increased in accordance with equation (I). The critical pressure ratio, however, will still hold and if the supercharge pressure were increased beyond the critical value the power output would remain the same. Thus at all speeds and throttle openings there will be an optimum maximum value for the supercharge or boost pressure.

The above discussion has been based on the assumption that the throttle valve in the carburettor acts as a well designed nozzle and this, of course, is not strictly true. There will, no doubt, be considerable friction losses in the flow over the valve and these losses will vary with the opening of the valve. Also, during the passage of the air through the throttle and along the manifold, the fuel, injected into the air stream at the throat of the choke, will be rapidly evaporating. This will cool the air considerably and tend to alter the total manifold pressure due to the partial pressure exerted by the vaporized fuel. A considerable amount of heat is supplied to the inlet manifold from the exhaust manifold and other hot engine parts so that the overall tendency is for the mixture in the inlet manifold to be at a higher temperature than the air supplied to the carburettor and thus not in accordance with the assumed adiabatic expansion over the throttle valve. However, all these effects depend on engine design, operating conditions and engine speed and could not be taken into account in a simple general equation. Furthermore, it has been assumed that the manifold pressure is a constant value at any particular combination of speed and throttle opening. Obviously, this is not the case as the manifold pressure fluctuates with the piston motion. The fluctuations depend on engine speed and the number of cylinders as well as the design of the manifold; therefore, to assume steady flow conditions will necessarily lead to inaccuracies in results calculated from the given equation. It may be pointed out, however, that with a large number of cylinders, four, six or eight, this approximation becomes less critical, whereas with a single cylinder engine large errors would, no doubt, be apparent.

With these conceptions in mind a comprehensive programme of tests was conducted to investigate the practical application of the theoretical laws to a standard four-cylinder vehicle engine.

Test engine and measurements

The engine used in the tests was a four-cylinder, four-stroke, side-valve engine of conventional design, incorporating a down-draught carburettor supplying an externally fitted induction manifold. The induction manifold had two passages, one at either end, feeding the two intake ports of the engine. The outer adjacent cylinders of the engine had common intake ports in the cylinder block. A "hot-spot" was provided at the centre of the manifold directly beneath the carburettor by the junction of the inlet and exhaust manifolds.

The test engine details were as follows :

Bore	2·4995 in
Stroke	3·5 in
Cubic Capacity	68·7 in³
Compression ratio	6·5
Valve Timing	Inlet opens T.D.C.; closes 46 deg A.B.D.C. Exhaust opens 50 deg B.B.D.C.; closes 20 deg. A.T.D.C.
Ignition Timing	Top dead centre at rest, with centrifugal control. Maximum advance, 8 deg.

The test engine was direct-coupled to a Heenan-Froude

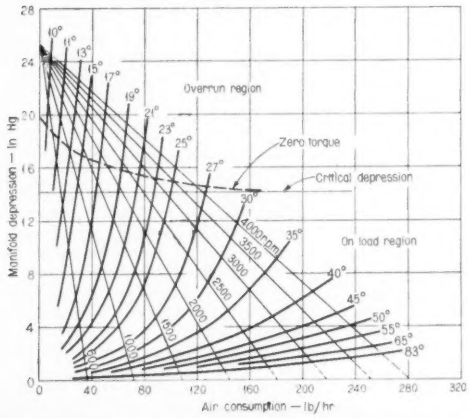


Fig. 3. Depression—Air relation. Constant speed and fixed throttle curves

Fig. 4. Relation between throttle area and angle of rotation

dynamometer for load tests; for overrun tests it was driven by a second engine of known output. The power input to the test engine on the overrun tests was calculated by a difference method. This input is designated by "negative torque" in the results—it includes friction as well as pumping losses.

Measurements taken on the test engine were :

(a) Torque, output or input, in lb-ft
(b) Fuel consumption in lb hr
(c) Air consumption in lb hr
(d) Manifold depression in centimetres of mercury
(e) Manifold stream temperatures in deg C.
(f) Throttle valve openings in degrees of rotation.

The fuel consumption was measured by means of a liquid flowmeter, correction being made for the temperature and viscosity of the fuel, Ref. 3. An Alcock viscous-flow air meter was used to determine the air consumption and, allowing for variations in atmospheric conditions, the results were expressed in pounds of air per hour.

The manifold depression was measured using a vertical mercury manometer connected by a tube through a large capacity bell jar to a tapping point at the centre of the induction manifold.

The stream temperature in the induction manifold was indicated at three points by unsheathed copper-constantin thermocouples situated in the centre and at each end of the manifold, at the junctions of the two end inlet ports in the cylinder block. The central thermocouple was placed directly above the hot-spot. Thermocouples were insulated from the manifold walls and were carefully calibrated in conjunction with a sensitive millivoltmeter before the commencement of the tests.

As the couples were unsheathed no account could be taken of the 'wet bulb' effect that might have arisen due to the deposition of liquid fuel droplets on the junctions. This factor constitutes a possible error in the temperature results presented; but it was considered that any other method that could have been used to indicate the temperature of the ingoing mixture would have given rise to even greater errors.

The setting of the throttle valve was indicated on a scale swept by a pointer connected to the valve spindle. This scale was calibrated in degrees of throttle opening and could be read accurately to within $\pm\frac{1}{4}$ deg. A scale reading of 10 deg corresponded to the closed position when the valve was against the slow running stop, previously set to idle the engine at 500 r.p.m. The maximum opening of the valve was represented by a scale reading of 83 deg, when the valve was against the other stop.

Tests were made at varying throttle openings from 10 deg to 83 deg with speeds varying from 500 r.p.m. to 4,000 r.p.m. including overrun conditions. Throughout all the tests the jacket cooling water at outlet was maintained at 80 deg C and the oil temperature between 75 and 80 deg C. Neither fan nor dynamo was fitted to the test engine.

In plotting the test results the variations in day-to-day ambient temperature were taken into account, although at no time was this variation more than 2 deg C.

Test results

Fig. 3 shows the results of manifold depression plotted against air consumption for tests at constant speeds. It will be noticed that the points lie on straight lines intersecting the manifold depression axis at 25.2 in Hg. Lines of constant throttle opening are also plotted and are shown as curves increasing in slope with rise in speed. The line of zero torque divides the regions between load conditions and overrun; and the points at which the constant throttle lines cut this curve indicate the engine speed on no-load at a particular throttle setting. The intercepts of the constant speed lines with the air consumption axis represent the air consumption of the engine if there was no inlet restrictions present. Calculations have shown, however, that these points are not equally spaced along the axis and do not represent the swept volume of the engine at the particular speeds. As the curves were drawn through so many points these plotted points have been omitted but in no instance was there any deviation from the straight lines drawn.

The relationship between the angular throttle opening and the projected area of the restriction is plotted in Fig. 4. This relationship was derived by analysing the dimensions of the throttle valve and the intake passage. At 10 deg the valve is very slightly open, as shown by the inset in Fig. 4. This slight opening was required in addition to the bleed air byepass provided in the carburettor, to idle the engine at 500 r.p.m.

It will be seen that beyond $75\frac{1}{2}$ deg the throttle area of opening remains constant as the obstruction due to the valve spindle limits any greater area of opening. It is considered, however, that although the throttle opening does not increase beyond $75\frac{1}{2}$ deg, there is a change in the flow characteristics over the valve as the angle is increased from $75\frac{1}{2}$ deg to 83 deg.

Fig. 5 indicates the relationship between the air consumption and the projected area of throttle opening, for constant speed tests. Only results of tests on load conditions have been plotted as the overrun tests would be indistinguishable from the zero torque line. Dotted lines show curves of constant manifold depression and these are almost straight above depressions of 5 in Hg. The constant speed curves tend to vertical asymptotes, especially at the lower speeds, representing the aspirated air consumption with no inlet restrictions. If the zero torque line were continued to cut the negative axis of throttle opening the area would represent the equivalent area of the bleed by-pass.

The variation in torque with manifold depression and air consumption is plotted in Figs. 6 and 7. At speeds below 1,500 r.p.m. the torque varies almost linearly with the air consumption under load conditions, but at speeds

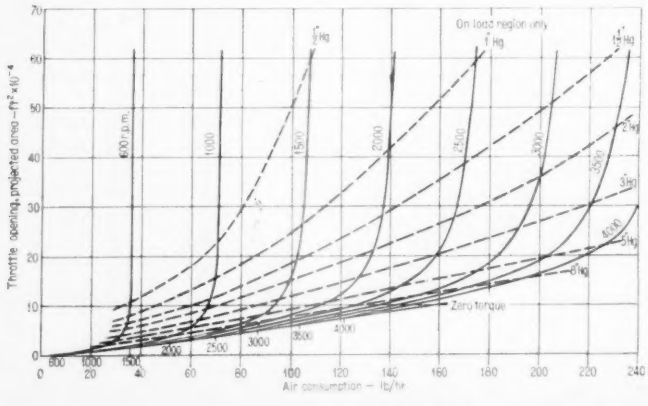


Fig. 5. Throttle opening—Air relation. Constant speed and manifold depression lines

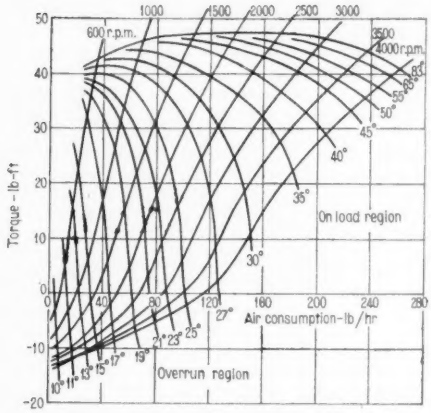


Fig. 6. Torque—Air relation. Constant speed and fixed throttle curves

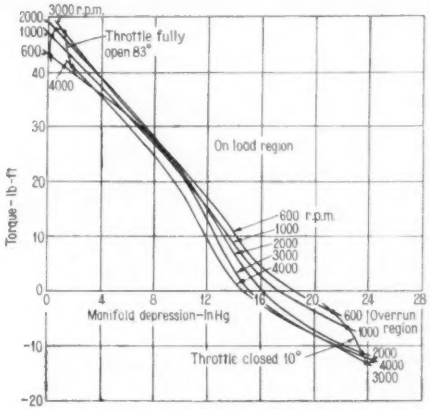


Fig. 7. Torque—Depression relation. Constant speed curves

above 1,500 r.p.m. there is a tendency for the curves to decrease in slope with increased air consumption. Above 35 deg the constant throttle lines tend to merge with the 83 deg line, Fig. 6, and at 1,500 r.p.m. there is no gain in torque between throttle openings 50 deg to 83 deg. In the overrun region the 4,000 r.p.m. constant speed line cuts the 3,500 r.p.m. line at about 14 deg throttle opening, which indicates that in Fig. 3 a 14 deg constant throttle line would be vertical between these speeds. At 10 deg throttle opening, Fig. 6, the air consumption is slightly less at 4,000 r.p.m. than at 3,000 r.p.m. and thus the 10 deg line in Fig. 3 will have a negative slope above 3,000 r.p.m.

In order to avoid confusion in Fig. 7 only the 83 deg and 10 deg constant throttle lines have been plotted. The 83 deg line is in the form of a loop and indicates a lower torque output at 4,000 r.p.m. than at 3,000 r.p.m. Although the manifold depression is greatest at 4,000 r.p.m. on the overrun with 10 deg throttle opening, the greatest resistance of the engine is shown to be at 3,000 r.p.m. The constant speed curves of Fig. 7 show that, over the lower range of manifold depressions, the depression corresponds approximately to a certain fixed torque irrespective of speed, but the inaccuracy of this assumption becomes greater on the overrun where the constant speed curves show greater divergence.

The results obtained from the measurements of the manifold stream temperature are plotted in Figs. 8-12. The effect of engine speed on the temperature of the mixture entering the cylinders is shown for the front and centre of the manifold in Figs. 8 and 9 respectively. Measurements at the rear end of the manifold yielded a figure similar to Fig. 8, with the temperatures, in general, 1 to 2 deg C less. At all three positions the highest temperatures were recorded in the overrun region; the temperatures increasing as the throttle was closed. At the ends of the manifold under load conditions the temperatures tended to decrease with speed until about 2,000 r.p.m. and then slightly increase with further increase in speed. This tendency increased at smaller throttle openings Although the temperature measured at the manifold centre, above the "hot-spot", varied in an essentially similar manner to the end temperatures, there was no regain in temperature at the higher speeds with throttle openings in excess of 40 deg. In all three positions, on

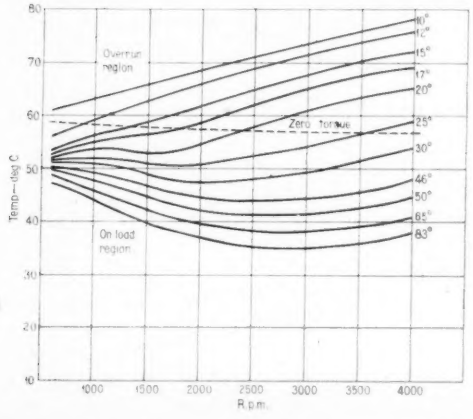


Fig. 8. Inlet manifold temperature. Fixed throttle curves, front end of manifold

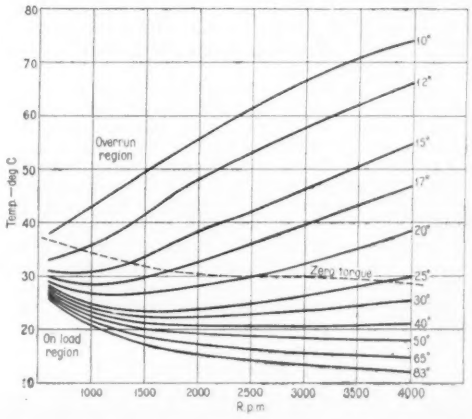


Fig. 9. Inlet manifold temperature. Fixed throttle curves, centre of manifold

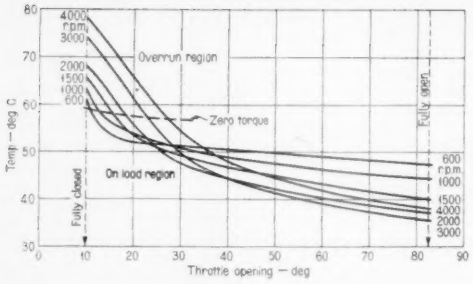


Fig. 10. Inlet manifold temperature. Constant r.p.m. curves, front end of manifold

the overrun the temperatures increased almost linearly with speed.

Figs. 10 and 11 indicate the variations in temperature with throttle opening at constant speeds. Again the highest temperatures were recorded in the overrun region at highest speeds and smallest throttle openings. On power the temperatures are lower, the greatest speed producing the lowest temperature. At no-load the temperature remained almost constant and if a zero speed line were drawn it would be represented by a horizontal straight line at atmospheric temperature.

The temperature gradient along the manifold is illustrated in Fig. 12 and this shows the lowest temperature, at all combinations of throttle setting and load, to be at the centre of the manifold directly above the "hot-spot". This may be partly due to the "wet bulb" effect mentioned earlier but it is doubtful whether this is entirely responsible for the marked temperature gradient along the manifold. The lines are similar for power and overrun conditions but on the overrun the temperatures are always higher than with the engine on load. The greatest temperature difference between the ends and the manifold centre was at 4,000 r.p.m. and 25 deg throttle opening on the overrun and the smallest difference was also on the overrun at 4,000 r.p.m. at 10 deg throttle opening. The centre-lines of the passages leading to the cylinder ports are indicated in Fig. 12 and it is seen that with the design of manifold used in these tests there exists only a few degrees difference in temperature between the mixture entering the front and rear cylinders under all operating conditions. It is reasonable, therefore, to suppose that the degree of vaporization of the fuel entering all the cylinders will be similar and that the mixture distribution will be fairly uniform with this type of induction manifold.

The fuel consumption has been plotted against speed in Fig. 13 and it is clearly shown that on the overrun there is no increase in consumption with speed at a particular throttle opening. This shows that if a vehicle were allowed to run down a steep hill with the engine being driven on the overrun and with the throttle closed, the fuel consumption would be that for idling the engine and would be independent of speed. It is, therefore, as economical to use the engine as a brake as to disengage the clutch and use the wheel brakes. Fig. 7 indicates the braking effect of the engine with closed throttle and it will be seen that the resistance increases with speed until 3,000 r.p.m., after which there is a slight decrease in braking effect.

Conclusions

An interesting comparison may be made between the results obtained from the theoretical equations presented at the beginning of this paper and the actual values obtained under test conditions. Certain features of the plotted curves can also be explained by analysing the results on a theoretical basis.

Equation (IV) may be used to estimate the intercept of the constant speed lines with the manifold depression axis, Fig. 3, assuming a suitable value of the index of expansion n can be obtained. Strictly speaking, as this equation is based on the assumption of no air flow there can be no combustion in the cylinder and, consequently, no heating of the exhaust products. Thus the air contained in the cylinder would be merely expanded and compressed by the piston motion and would presumably remain at a temperature near to that of the jacket cooling water. If the expansion be assumed isothermal ($n = 1$), the manifold depression is calculated as 24.44 in Hg. (assuming atmospheric pressure 30 in Hg.). This calculated value compares favourably with the actual intercept of 25.2 in Hg. obtained from the curves. Using a value of $n = 1.07$ gives the actual intercept of 25.2 and thus it would appear that during the expansion process a very small amount of heat is rejected to the jackets. However, for practical purposes the value $n = 1$ would give a reasonable estimation of this intercept.

The critical manifold depression of 14.16 in Hg. has been indicated on Fig. 3 and it will be seen that at higher speeds the zero torque line appears to be assymptotic to this critical value. At the lower speeds, however, the manifold depression is inclined to exceed this value. These discrepancies are due to the divergence from the theoretical assumptions at low speeds. The air flow across the valve will no doubt depart greatly from the assumed adiabatic conditions and the effects of the pressure fluctuations will become more serious.

The constant throttle opening lines in Fig. 3 increase in slope as they approach the critical manifold depression and at the higher speeds, where the zero torque line approaches the critical depression, the constant throttle lines become almost vertical thus indicating that when running on no-load the engine is receiving the maximum possible air supply through the carburettor at the particular throttle valve setting. There can be no further increase in speed of the engine on no-load at a given throttle position when the manifold depression has reached the critical value, as further speed increase would reduce the manifold depression below the critical value and thus by equation (v) the air supply and hence the fuel supply would remain constant, and in consequence the power output of the engine to be used in overcoming the frictional resistance would not increase.

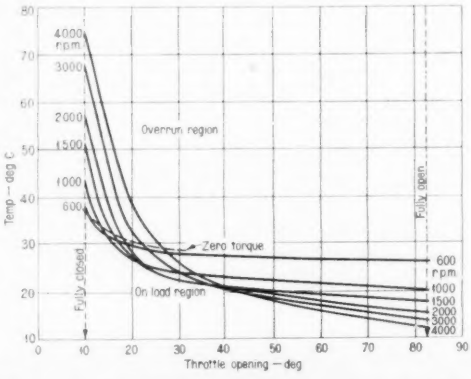


Fig. 11. Inlet manifold temperature. Constant r.p.m. curves, centre of manifold

As previously mentioned in connection with Fig. 6, a constant throttle line of 14 deg opening in Fig. 3 would be vertical between 3,500 r.p.m. and 4,000 r.p.m. and the 10 deg opening throttle line assumes a negative slope between these speeds thus indicating no increase in air consumption with increase in speed at constant throttle opening on the overrun.

Substituting the appropriate values of P_A and T_A in equation (VI), the maximum air flow for a given throttle opening may be written as

$$W_{Max} = 1.748\, C_D A_T \times 10^5 \text{ lb/hour.}$$

If it be assumed that $C_D = 0.90$ for all angles of throttle opening then

$$W_{Max} = 1.575\, A_T \times 10^5 \text{ lb/hour.}$$

This equation may be plotted on Fig. 5 and will be found to be almost on top of the zero torque line. (A slight discrepancy will arise as the value A_T does not include the bleed air by-pass.)

If equation (V) is used (with values of manifold depression below the critical value) linear relationships will be obtained between throttle opening and air consumption for fixed manifold depressions. Fig. 5 shows these relationships to be obeyed above a depression of 5 in Hg. but below this value the practical results no longer obey these linear laws.

Equation (II) suggests the theoretical variation in fuel consumption under no-load conditions. It has already been suggested that theoretically on no-load, the manifold depression will remain constant at the critical value so that equation (II) may be written as

$$w = \frac{\text{r.p.m.} \times k}{\xi} (K + F)$$

where K and k are constants. Above a speed of 1,500 r.p.m. Fig. 13, the no-load fuel consumption varies linearly with the speed so that it would appear that the combustion efficiency and the frictional resistance of the engine on no-load are independent of engine speed. In Fig. 3 below a speed of 1,500 r.p.m. there is a wide divergence between the critical depression and the zero torque so that above relation does not hold.

Taking a particular throttle opening, say 25 deg, the theoretical air consumption on no-load (i.e. when the manifold depression is the critical value) is obtained from equation (VI) as 100.9 lb/hour, whereas the actual value is read from Fig. 3 at zero torque as 101 lb/hour.

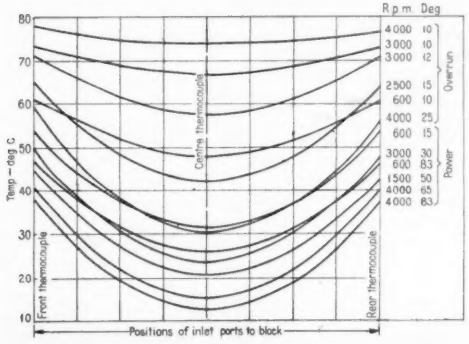


Fig. 12. Inlet manifold temperature distribution

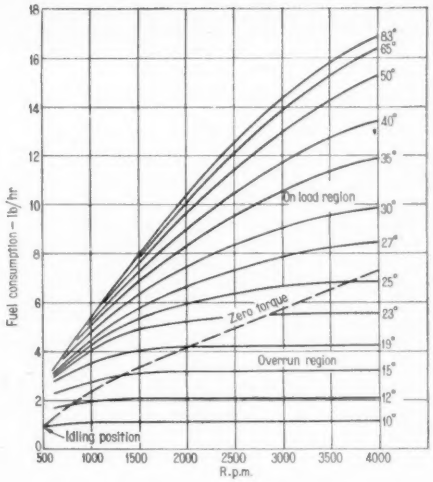


Fig. 13. Fuel consumption

The engine speed on no-load at 25 deg throttle opening may be calculated from equation (III) using the theoretical value of the air consumption. This gives a speed of 3,340 r.p.m. compared with the actual value of 3,400 r.p.m. Alternatively, equation (III) and (V) or (VI) may be combined to estimate the manifold depression produced by any combination of throttle setting and speed when the engine is on load, no-load or overrun.

Equations (I) and (II) can only be used to estimate the power output and fuel consumption of the engine if the combustion efficiency of the engine is known throughout the range of working conditions. This value will depend upon the cylinder head design, the mixture strength and ignition timing as well as many other factors associated with the combustion process, and consequently the value of these equations is limited to cases where such factors are known. However, in order to gain an approximate estimation of the engine performance a value of the combustion efficiency may be assumed and the value of w_F, the fuel consumption, may be estimated from a knowledge of the air-fuel ratio and the air consumption calculated from equations (III), (V) or (VI).

In conclusion it is submitted that, whereas the theoretical equations presented cannot be expected to provide extremely accurate estimations of the complex quantities concerned with engine performance, they do permit calculations to be made that will produce reasonably accurate results for design and investigation purposes.

The work described above was carried out in the mechanical engineering laboratories of Queen Mary College, University of London, and the authors are indebted to Professor E. Giffen for suggestions and help throughout the work.

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A POWER BRAKE UNIT

The Bendix Treadle-Vac for the Packard Easamatic System

FOR many years, power braking systems have been installed in heavy commercial vehicles. However, the Packard Motor Car Co. is probably the first to apply this principle, combining the hydraulic master cylinder with the pneumatic servo unit, to passenger cars manufactured in large quantities. The modern trend in America, in addition to the simplification of control from the driver's point of view, is to reduce the amount of physical effort needed when driving the car. It is to this end that the new unit has been designed.

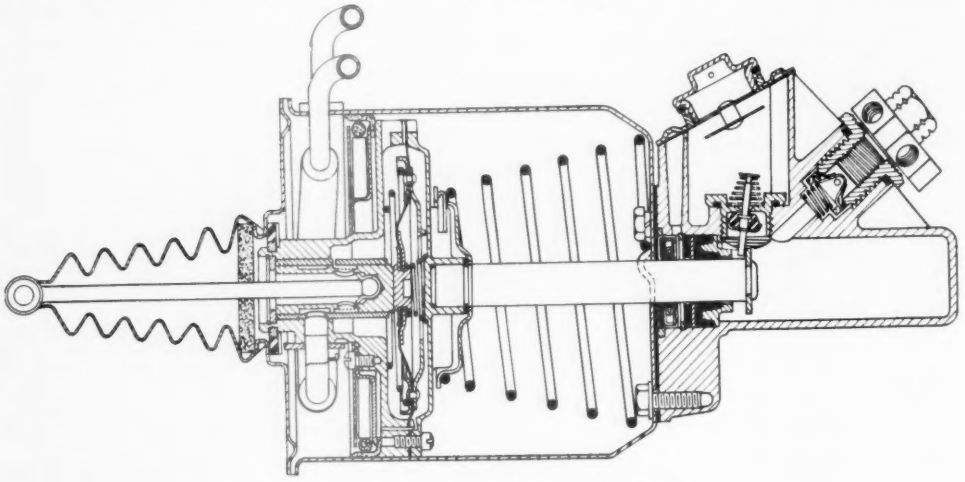
It might be argued that servo-operation is essential in large cars. This is confirmed, to some extent, by experience in this country with custom-built limousines. Owing to the greater leverage necessary to operate the braking system of the heavier vehicle, a rather long pedal travel must be employed. The Bendix system overcomes this difficulty, and the brake pedal need then project, from the toe-board, only slightly higher than the throttle pedal. This will appreciably reduce the time necessary to apply the brakes. The manufacturers state that this time is lessened by a quarter. Moreover, the foot may be moved from one control to the other simply by pivoting on the heel and, so far as ease of operation is concerned, the Treadle-Vac is comparable with the throttle control. Pedal operation is, no doubt, strange at first, but there is no reason why drivers should not quickly become accustomed to it. As the pedal is depressed, the resistance is increased progressively to give what is claimed to be an adequate feel of the control.

The manufacturers state that the Treadle-Vac unit does not add to the complication of the brake control, but replaces the ordinary hydraulic system, and requires no more attention than the conventional master cylinder. Reliability has been ensured by using the well-proved principles of the Bendix Hydrovac system, and by the usual development trials.

Air intake depression supplies power for the Treadle-Vac unit. Adjacent to the induction manifold, in the pipeline to the servo unit is a check valve, and between this valve and the servo is a T-junction to which is connected a reserve tank. This tank is incorporated so that, should the engine stall, the brakes may still be operated. The end of the pipeline connecting the pneumatic cylinder to atmosphere is fitted with an air cleaner mounted on the dash. To further ensure that the air entering is as clean as possible, it is taken from inside the saloon.

The brake control unit consists of three parts; a hydraulic master cylinder, a pneumatic power cylinder and a mechanically actuated control valve. Incorporated in the casting that forms the hydraulic cylinder is a fluid reservoir with its compensator, or recuperator valve, and a residual check valve. Co-axial with the master cylinder is a plunger which is formed by the rod of the pneumatic piston. Around the plunger and housed in the outer end of the hydraulic master cylinder are two seals. The inner, or front one, made of rubber, is of channel section with its flanges, bearing on the plunger and the housing, directed into the cylinder. It is held in position by a flanged washer. The flange seats on a shoulder in the casting, and a spigot projection around the washer fits into the channel of the seal. Bearing against the other face of the seal is a Spauldite washer that forms a guide for the plunger. The whole assembly is retained in position by a plain washer, and a Truarc, or Spirolex, circlip in the housing. On the other side of the circlip is a lip-type leather oil seal. This is retained by the end face of the bolted-on pneumatic cylinder. Should any oil escape past the inner seal, it is carried through a vertical duct from between the two seals to a point, above the oil level, in the hydraulic reservoir.

Secured, together with a joint washer, to the top of the reservoir by six cheese-headed screws, is a cover plate on which is a boss for the filler plug. In the base of the reservoir is the compensator valve. This is



General arrangement of the Bendix Treadle-Vac unit

mounted in a screwed-in housing furnished with an axially drilled hole. The valve itself consists of a spindle, positioned vertically in the hole and having, approximately mid-way between its two ends, a flange on which is mounted the rubber valve head. The flange and head are positioned below the hole, and the assembly is retained by a conical coil spring around the spindle above the hole. At its upper end this spring bears against a C-shaped washer sprung into a groove around the end of the spindle, while its larger, lower end seats in a circular recess in the valve body. When the brakes are released, a washer, peened in position around the end of the hydraulic plunger, strikes the downward projecting end of the spindle. This tilts the valve, ensures recuperation is complete and, perhaps more important, prevents damage occurring as the result of expansion of the fluid arising from changes of atmospheric temperature.

When the plunger is operated to apply the brakes, fluid is forced out of the cylinder past the residual check valve. The purpose of this valve is to ensure, even if there should be a slight air leak in the brake system, that the pipeline shall remain full of fluid. The hollow cylindrical check valve body is screwed into the fluid outlet. In the inner end of the body is engaged a cone-shaped rubber valve mounted on a similarly shaped metal retainer. These components are held in position by a light compression spring.

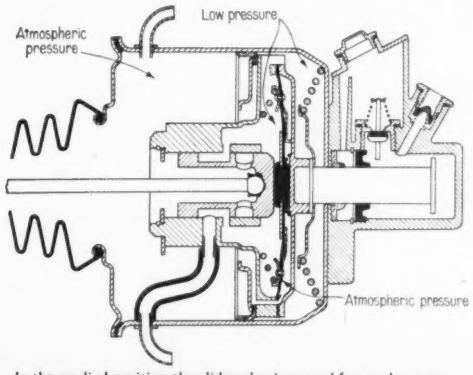
The assembly operates in the following manner: When the hydraulic plunger is forced into the cylinder, fluid passes through holes in the walls of the retainer cone, and then, through a hole in the apex of the rubber cone, to the brake pipeline. When the plunger is released, the pressure in the pipeline unseats the cone-shaped valve, and the fluid passes around it back to the cylinder. The compression spring reseats the valve as soon as the pressure in the pipeline falls. No more fluid can then pass the valve as the rubber cone covers the holes in the retainer.

The pneumatic cylinder is a pressing, and is secured to the master cylinder by three set bolts locked by spring washers. A Vellumoid joint washer is interposed between the base of the pneumatic cylinder and the master cylinder. Inside, and bearing against the base, is the large end of the conical piston-return spring. The small end bears against a dished circular plate. Seated in this is a U-shaped washer that registers in a groove around the hydraulic plunger to take the thrust of the spring. The two ends of the wire forming the spring are each bent inwards; that at the front registers between the head of one of the three retaining bolts and a projection pressed into the end of the cylinder, while the other passes through a slot in the edge of the retaining plate, and engages between radial vanes integral with the rear face of the diaphragm cover. This arrangement locates the piston against rotation so that it may more effectively bed into the cylinder.

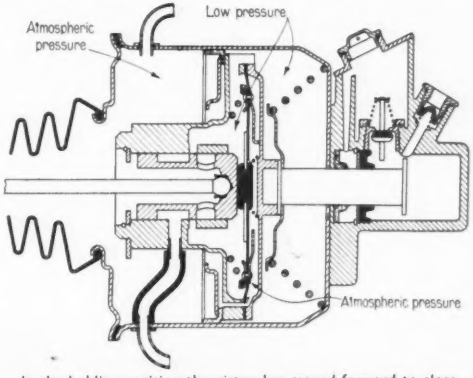
There are three main parts to the pneumatic piston. These are the diaphragm assembly, the piston seal assembly, and the valve mechanism. At the front of the diaphragm assembly is a die-cast cover that forms the front face of the piston. The end of the hydraulic plunger is carried in a boss on this cover, and it is held there by the piston-return spring. The cover is dish-shaped, and radially disposed inside it are five triangular-shaped webs to prevent the flexible diaphragm from bottoming and sticking in the manner of a suction-cup. The diaphragm is clamped, around its periphery, between its cover and the piston plate. This plate is also dished, so that between it and the diaphragm cover there is a fairly large space to house the return coil-spring for the diaphragm, as well as that for the slide valve. Both these springs are conical, as in this form they may be compressed to a relatively small length without becoming coil bound.

The diaphragm is a rubberized disc. At its centre is a circular hole reinforced by a metal washer on each side, the two washers being riveted together through the diaphragm. Two more metal rings are riveted together in a similar manner nearer the outer periphery. One of these rings protects the diaphragm against chafing on the webs in its cover, while that on the other side seats its return spring. These outer rings reduce the tendency of the diaphragm, when under pressure, to be pulled out from the flanges of the two covers between which it is clamped. Pressed into the central hole is a thick, cup-shaped, rubber stop. Bearing against the rear face of the stop is the end of the slide valve, and against the front face is the slide-valve return spring. The other end of this spring is located around a small boss on the diaphragm cover.

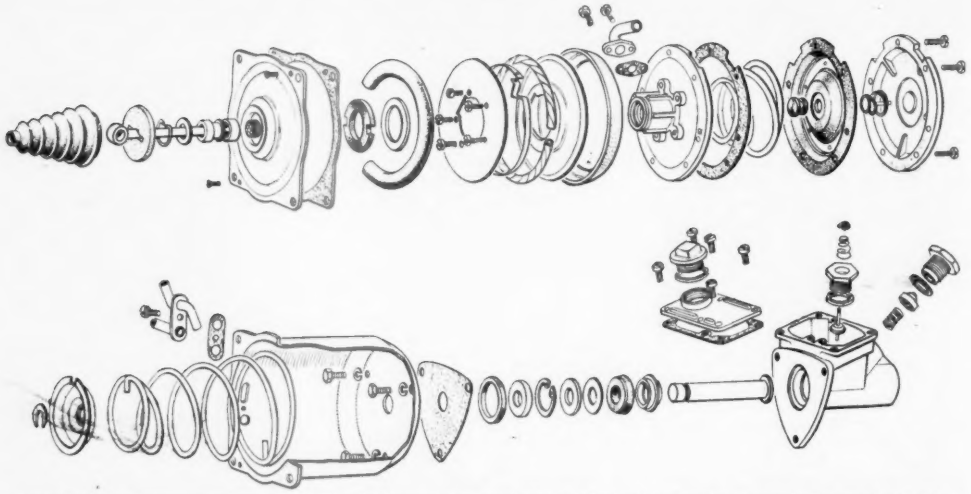
Integral with the die-cast piston plate, and projecting to the rear, is a sleeve that houses the slide valve assembly. Around this sleeve are cast-on ribs that support a stop washer at the rear, and a small boss to which is fitted, together with a joint washer, an adaptor for the flexible, induction-pressure pipe. The adaptor is secured by two set screws. The flexible pipe is coiled almost through 360 deg around the inside of the cylinder, and it is connected at its other end to an adaptor secured, by one centrally positioned screw, to the cylinder side wall. This adaptor also incorporates a connection for the pipe from the air filter to the cylinder, a joint washer being used under the flange of the adaptor to prevent entry of dust.



In the applied position the slide valve is moved forward to open the transfer ports



In the holding position the piston has moved forward to close the transfer ports



Although there are many components in the Treadle-Vac unit, they are of simple design, and inexpensive to manufacture on a large scale

Also positioned around the sleeve is the pneumatic piston seal assembly. This consists of a large, leather cup-washer held against the piston plate by a retainer plate. Around the inside of the cup-washer is a wick that is soaked with oil to keep the leather in a soft condition. This wick is held in position by a circular expander spring, which also serves to keep the sides of the cup-washer pressed against the wall of the cylinder. The whole assembly is held together by another retainer plate secured to the piston plate by five set screws.

A liner, pressed into the sleeve on the pneumatic piston plate, houses the piston-type slide valve. Seated in a spherical pressed-steel cup, in the inner end of the valve bore, is the ball end of the push rod. An outwardly turned lip around the rim of the cup is constrained against the bore of the valve to prevent the ball end of the push rod from sliding out. At the other end of the rod is an eye for connection to the brake pedal. Above this connection, the pedal is pivot-mounted to the toe-board. Around the outer periphery of the valve there are two annular grooves. The one at the rear acts as a transfer passage for the induction pipe pressure, while the other serves a similar purpose, but for atmospheric pressure. Holes are drilled radially in this latter groove to maintain a constant atmospheric pressure to the rear of the valve and in the rubber gaiter that is fitted around the push rod. A wire gauze screen in the groove gives additional protection from dust. The valve is retained in the sleeve by a washer and a circlip, fitted in the rear end of the boss.

Around the shouldered outer periphery of this boss is the steel stop washer, already mentioned. Sprung into a groove behind this is a rubber stop washer that, when the brakes are off, comes up against the pressed steel end cover of the pneumatic cylinder. Two slots, diametrically opposite each other on the outer periphery of the rubber washer, prevent sticking. The cover is approximately rectangular in shape, and so is the flange around the cylinder, to which it is secured by two countersunk set screws. These are positioned diagonally opposite one another. Adequate sealing, against the ingress of dust, is ensured by interposing a joint washer between the two faces, and by the clamping action of the four bolts that pass through the corners of the flanges to secure the unit to the toe-board. A lip is turned out around the hole in the cover through which the operating rod passes. Sprung over the lip is the rim of the rubber gaiter. Housed in this end of the gaiter is a disc-shaped felt seal providing a further safeguard against dust entering the unit.

Operation of the Bendix Treadle-Vac unit is readily understood if it is remembered that, at all times, the pressure behind the pneumatic piston is that of the atmosphere outside. This pressure is also transferred, through drillings in the flange of the piston plate and the diaphragm cover, to the space between the diaphragm and its cover. When the brake pedal is depressed, it moves the slide valve forwards, and the air intake depression is communicated, through the transfer passage around the valve, to the space between the diaphragm and the piston plate. Furthermore, ducts in the flange of the piston plate aligned with slots in the flange of the diaphragm, communicate between this space and the cylinder in front of the pneumatic piston. As a result, two actions take place. The atmospheric pressure in front of the diaphragm causes it to move to the rear and apply a feel-back through the slide valve, push rod and the pedal control. At the same time, the pressure at the rear of the pneumatic piston pushes it forward, together with the hydraulic plunger which operates the brakes. Should the pneumatic system fail for any reason, direct application is, of course, possible.

In the following way, braking is made proportional to the degree of pedal travel. If the brake pedal is only depressed a short distance, the pneumatic piston, carrying with it the sleeve in which is the slide valve, automatically closes the low pressure port when it has moved forward a corresponding distance in the cylinder. Feel-back also depends on pedal travel. This is because it is governed by the sub-atmospheric pressure which is, of course, the same behind the diaphragm as in front of the piston, and which is proportional to the hydraulic pressure, in the master cylinder.

When the brake pedal is released, the slide valve is moved to the rear by its return spring, thus closing the low pressure port and opening the atmospheric port to the low pressure chambers. The piston is then returned by its spring as well as by the hydraulic pressure in the brake system. At the end of the return stroke, the washer on the hydraulic plunger unseats the compensating valve, and releases all pressure in the master cylinder.

ENGINE MANUFACTURE

Recent Production Developments at the Works of Vauxhall Motors Ltd.

(*Continued from page* 267.)

PREVIOUS articles in this series described the machines and methods used by Vauxhall Motors Ltd. in the manufacture of cylinder blocks and cylinder heads for the new Velox and Wyvern engines. For both these components, the machining lines were laid out to allow four- and six-cylinder engine components to be machined in any desired sequence without there being necessity for any tooling changes. This, the concluding article of the series, deals with the machines and methods for three components common to both engines, the connecting rod, the piston and the water pump body.

Connecting rod production

Separate stampings are used for the connecting rods and the caps. The chief points of interest in the machining of these components are, first, the inclusion of operations in the machining sequence that are additional to the sequence normally employed in the production of automobile connecting rods, and second, the use where possible of machines that allow several components to be machined simultaneously. The additional operations have been included in the sequence with a view to eliminating the necessity for a setting operation on the assembled rod after all the machining operations have been completed. While it would not be true to say that there has been complete elimination of the necessity for setting, there is evidence that the additional operations have at least greatly reduced the number of rods that have to be set.

On the connecting rod stamping, the first machining operation is carried out on an Archdale fixed-centre type multi-spindle drilling machine with hydraulic feed. It is tooled for drilling, rough and finish reaming the small end hole to give location for subsequent operations. The machine has 12 spindles, carrying four drills, four rough reamers and four finishing reamers. A four-station, power-operated work fixture is mounted on the machine table. Each station carries work-holding fixtures for four rods. Loading and location are simply effected. The rod is placed in an approximate position with the small end boss lying within vee jaws. Simple movement of a lever actuates a cam to advance a contoured pad into the large end half-bore and push the small end boss into contact with the vee jaws. In short, movement of the lever both locates and clamps the component. The set-up is shown in Fig. 46.

An automatic cycle is initiated by means of a push button control. The cycle is:—

(1) Table indexes one station.
(2) Head feeds down the pre-set depth.
(3) Head retracts to out position.
(4) Spindles stop.

As it is possible to unload and reload the fixtures at the idle station while the work is proceeding at the other three stations, there is practically continuous production from this machine. The operation is completed on four rods for each cycle.

A Weatherley Oilgear double-slide vertical hydraulic surface broaching machine is used for the next operation. The shuttle table carries two identical two-station fixtures, and the broaches on each slide carry out the functions. From the first position the half-bore is broached to leave an allowance for semi-finish and fine boring operations, and at the same time the sides of the bolt bosses are broached. The component is then transferred to the second position in the fixture for broaching the joint face and the back of the bolt bosses. An allowance of 0·005/0·010 in is left on the joint face for removal at a later grinding operation.

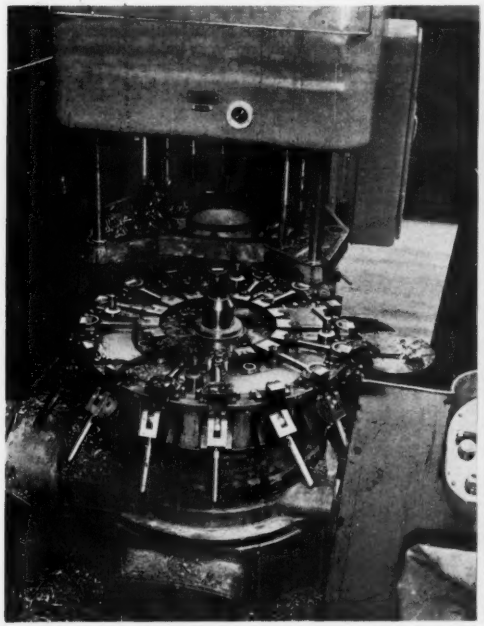


Fig. 46. Archdale 12-spindle, 4-station machine for drilling and reaming connecting rod small ends

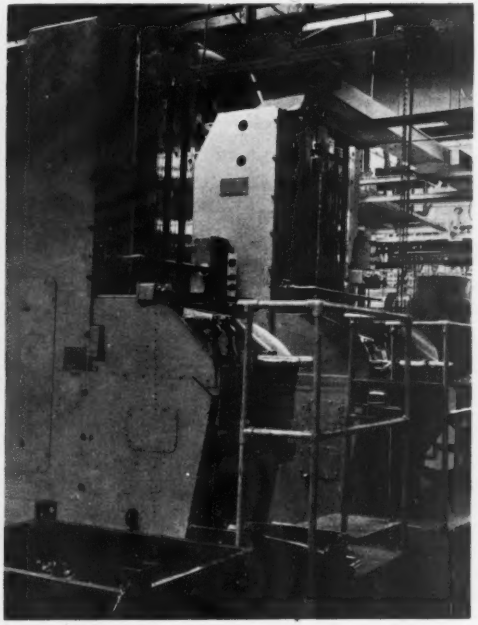


Fig. 47. Weatherley Oilgear hydraulic surface broaching machine for connecting rod caps

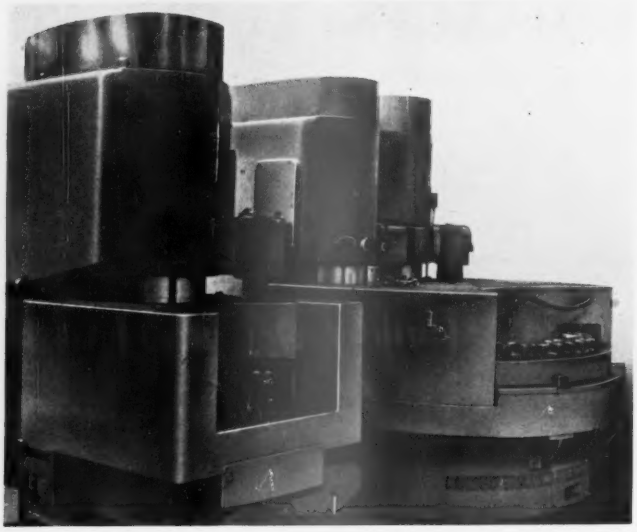


Fig. 48. Lumsden continuous surface grinders. The machine on the right grinds side faces, and that on the left, the joint faces of connecting rods and caps

Meanwhile, surface broaching operations are being carried out on the connecting rod caps. Two Weatherley Oilgear surface broaching machines are used. They are shown in Fig. 47. The first is a single slide machine carrying broaches for machining the sides of the bolt bosses of two caps at every pass. From this machine, the caps are transferred to a double-slide machine with two identical two-station fixtures on the shuttle table. The broaches on each slide perform the same work. At the first station, the half-bore is broached to leave an allowance for subsequent machining operations, while the joint faces and the bolt bosses are broached at the second station. Before any further machining is carried out, the broached rods and caps are washed and then shot-peened on the bolt bosses.

Both caps and rods are then transferred to a Lumsden double-column continuous surface grinding machine, shown at the right of Fig. 48. On this machine both sides of the large and small ends of the rod and both sides of the cap are ground to width, leaving an allowance for finish grinding after the cap has been assembled to the rod. The machine has a large diameter table carrying fixtures for rods and caps alternately in pairs. Of each pair of work fixtures, one is for the first grind on one side, while the other is for grinding the second side. Therefore, each rod and cap makes two passes on this machine.

The loading position of the double column Lumsden machine is shown in Fig. 49. Clamping is effected automatically. The outer jaw of each work fixture is actuated by a cam which is applied automatically before the work approaches the first wheel. It is later automatically released after the work has passed under the second wheel. In addition to incorporating automatic application and release of the clamps, this machine also has an electronic sizing device to each spindle. As wheel wear occurs, this device automatically causes the wheel to feed down by a compensatory amount. The sensitivity of this device is in the order of 0·0005 in.

A single spindle Lumsden surface grinder is used at the next operation to finish grind simultaneously the joint faces on the rod and the cap. Each of the several fixtures on the rotary table is designed to take one rod and one cap. Location is taken from one of the faces ground on the double column machine. This ensures that the side faces and the joint faces are truly at right angles. As with the previous machine, this machine incorporates a sizing device that automatically provides down-feed of the wheel to compensate for wheel wear. The loading station is shown in Fig. 50.

Separate machines are used for the rod and the cap at the next operation. Each is an Archdale cam-feed unit horizontal machine with a two-spindle attachment tooled for counterboring the bolt bosses. From these machines the parts come together again to an Archdale vertical, hydraulic feed multi-spindle drilling machine, see Fig. 51. A four-station rotary indexing fixture is mounted on this machine. Each station has work-holding fixtures for four components, either rods or caps. The usual loading is to have two rods and two caps at each station. There are three working stations, one for drilling, one for rough reaming and one for finish reaming the bolt holes. The table is power indexed and there is an automatic machine cycle, comprising table index, down-feed of head, retraction of head and stop. As unloading and reloading the fixtures can be carried out during the working cycle, there is practically continuous production.

Another Archdale fixed-centre type vertical multi-spindle drilling machine is used for the next major operation. There is a four-station indexing fixture with two work-holding fixtures at each station. Of each pair of work fixtures, one is designed to hold a rod with the

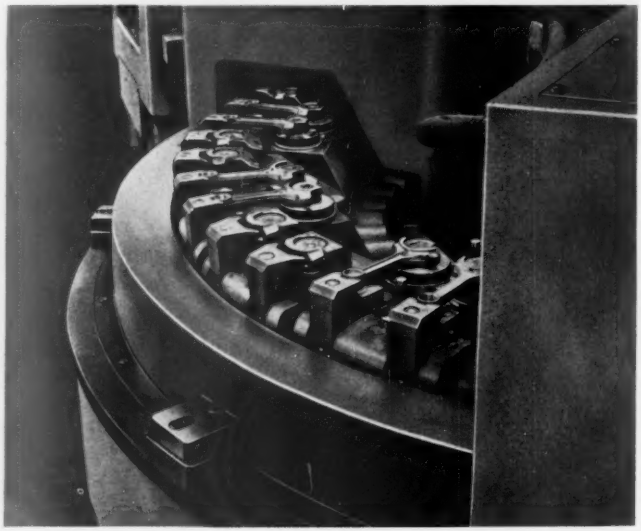


Fig. 49. The loading station for the Lumsden double-column machine shown in Fig. 48

small end in the working position while the other presents the large end half bore to the tools. At the first working station on this machine, a single drill drills a $\frac{3}{32}$ in hole part through the large end. There are two drills at the second station. One drills through at $\frac{1}{16}$ in diameter the $\frac{3}{32}$ in hole part-drilled at the first station, while the other drills an oil hole in the small end of the connecting rod in the other work fixture. Finally, at the third station, the hole drilled at the first station is chamfered. The set-up is illustrated in Fig. 52.

Lock tag slots are then milled in the half bores of the rod and cap, following which the small end bore is finish reamed to size for taking the bushing. The bolt holes are then line reamed and the cap is assembled to the rod. The assembled rod is then transferred to an Archdale single spindle machine to have both sides of the large end bore chamfered.

A final grinding operation for width is then carried out on both the large and small ends. The machine is a Lumsden double column surface grinder which is similar in all respects to the machine used for the first grinding on width, except that instead of alternate pairs of work fixtures for rods and caps the work fixtures are in identical pairs. For the first pass, the work is mounted in one fixture of a pair while one side is ground. It is then turned over and transferred to the second fixture of the pair for grinding the other side.

At first sight it may appear that this final grinding for width is unnecessary, but it does produce very accurate datum faces to act as locations for the final operations on the large and small bores. It thus helps to ensure that the axes of the two bores are in the correct relationship, that is, they are parallel and lie in the same plane.

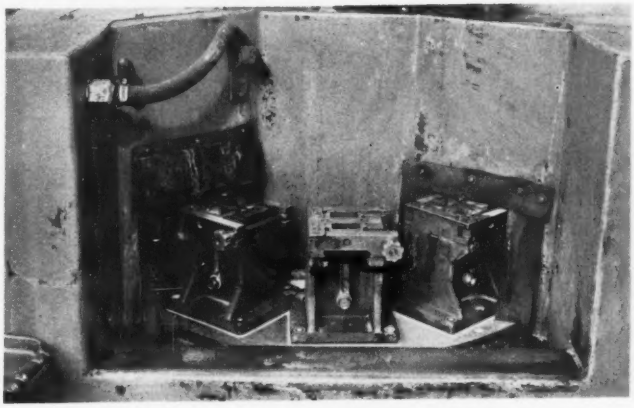


Fig. 50. The loading station for the Lumsden machine for grinding connecting rod and cap joint faces

As the first stage in finishing the bores, the large end is reamed on an Archdale vertical machine, following which the bush is pressed into the small end. Two further operations complete the machining. At the first of these a Precimax double-end fine borer is used to finish the small end bore to size and to fine bore the large end to a dimension that leaves an allowance for final sizing on a Micromatic Microhone machine, which not only automatically produces the correct size but also gives a remarkably high degree of surface finish. The finished rod is then de-magnetized, washed and given a complete inspection. Finally, before the rods are transferred to the engine assembly line, they are weighed and graded into sets. In any one set, the weight variation for both the large and small ends must not exceed $\frac{1}{4}$ oz.

Piston production

Whereas in the production of connecting rods, Vauxhall Motors Ltd. in the interests of increased accuracy have introduced operations not generally included in the machining sequence for automobile connecting rods, in the production of pistons they have

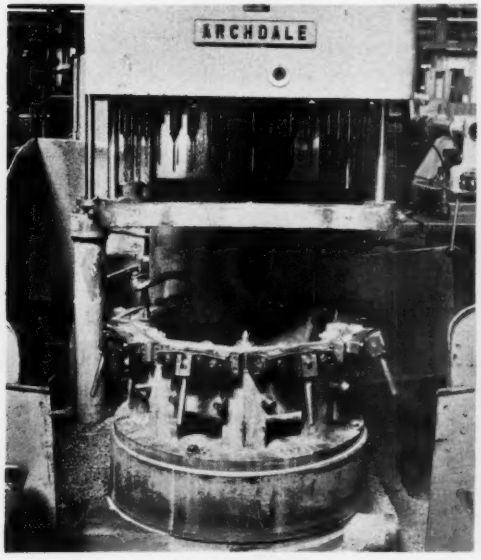


Fig. 51. Archdale four-station machine for drilling, rough and finish reaming bolt holes in connecting rods and caps

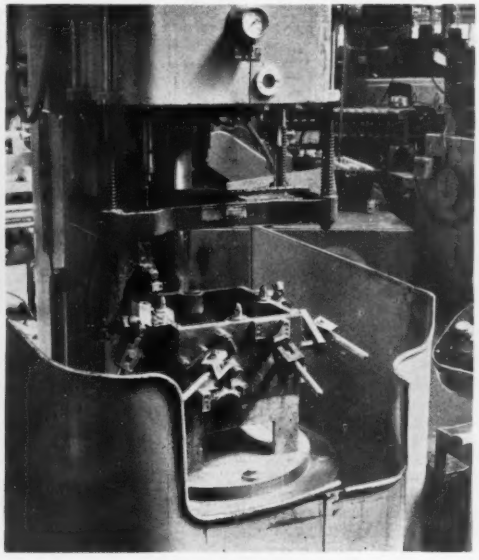


Fig. 52. Archdale machine for drilling oil holes in the small and large ends of Vauxhall connecting rods

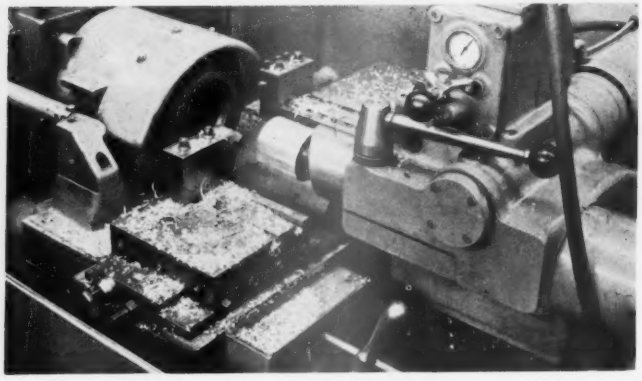


Fig. 53. Boring and facing the piston skirt on a Maximinor lathe

setting a centre is drilled in the pip on the crown, while from the front and rear tool posts the register diameter is bored, and the skirt end is faced to length and chamfered. This machine is shown in Fig. 53.

An Archdale two-way multi-drilling machine specially developed for machining pistons is used for the next operation. It has a five-station trunnion fixture and each station is arranged to take two castings. This machine is illustrated in Fig. 54. Loading and unloading are carried out at the first station. At the second station the gudgeon pin holes are core drilled. Out-feed tools are mounted at the third station to form the circlip grooves. At the fourth station the guidgeon pin holes are semi-finish reamed, and at the fifth they are chamfered. There is simultaneous machining on both castings at each

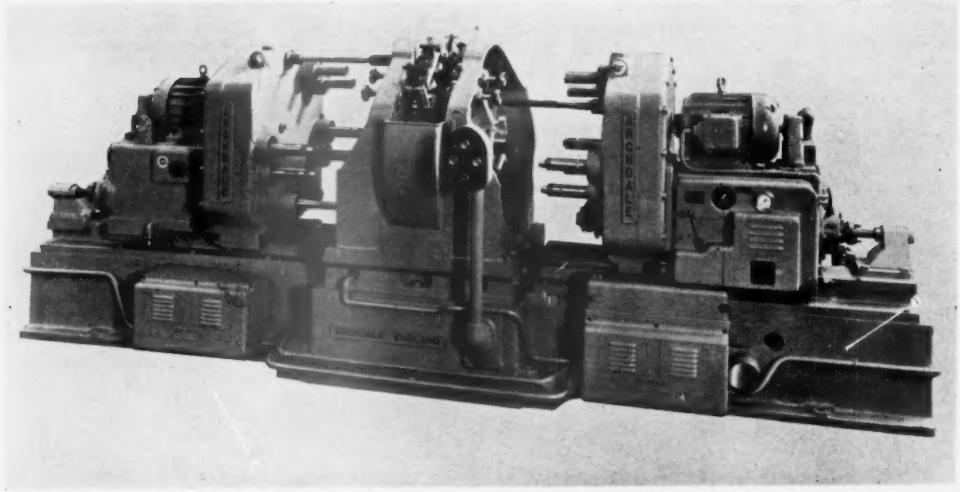


Fig. 54. Archdale 5-way machine for drilling, semi-finish reaming and chamfering gudgeon pin holes and forming circlip grooves

omitted some operations that are normal practice. For example, there is only one grinding operation, and that confined to the skirt. The land diameters are not ground at all, but are only fine turned to size. It should be stressed that this practice does not in any way affect engine efficiency, although it simplifies piston production.

An aluminium alloy piston is used. As cast, it has central pip on the crown. The first machining operation is to rough turn the outer diameter on a Herbert 5B capstan. For the second operation the casting is transferred to a Mamiminor lathe which has a special centring attachment through the spindle and a special air-operated tailstock. The component is loaded on to a locating device on the tailstock, which is then advanced to push the casting into a pot type chuck. At this

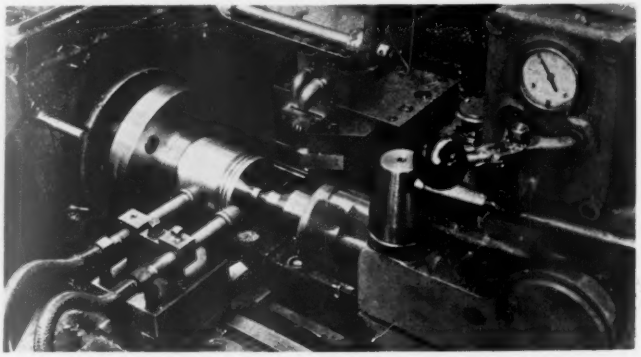


Fig. 55. Finish turning skirt and land diameters, forming ring grooves and facing the piston crown

Fig. 56. Special purpose Pollard machine for drilling piston oil holes

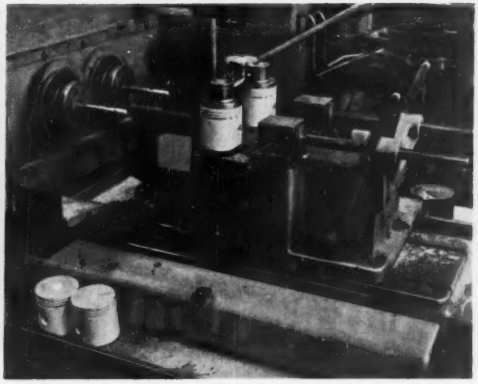


Fig. 57. Set-up for finishing gudgeon pin holes on a Precimax fine borer

station. Standard Archdale self-contained hydraulic feed units are used on this machine which has an automatic cycle.

From the Archdale machine the casting is transferred to a Maximinor lathe, see Fig. 55, which is tooled for turning the land and skirt diameters, forming the ring grooves—two compression and one scraper—and finish facing and chamfering the crown. The component is mounted on a special turning fixture which incorporates a drawbar and clamp pin. The tailstock centre registers in the centre in the crown.

Turning is effected from two tools in the front tool post, one for the skirt diameter and the other for the land diameters. Close dimensional accuracy and a high degree of surface finish must be maintained. Three magazine-mounted grooving tools, three groove chamfering tools also magazine mounted, and tools for facing and chamfering the crown are mounted in the rear tool post. In-feed for the crown facing tool is limited to leave the pip with its centre for use at the next operation.

The piston design calls for ovality, taper and a barrel form on the skirt. For producing these a Scrivener special piston grinding machine is employed. The piston is mounted on a work spindle carried in a housing that is in its turn mounted on a cam actuated slide. The cam is designed to give the piston an oscillating movement in relation to the grinding wheel of such a character that the specified form is automatically generated.

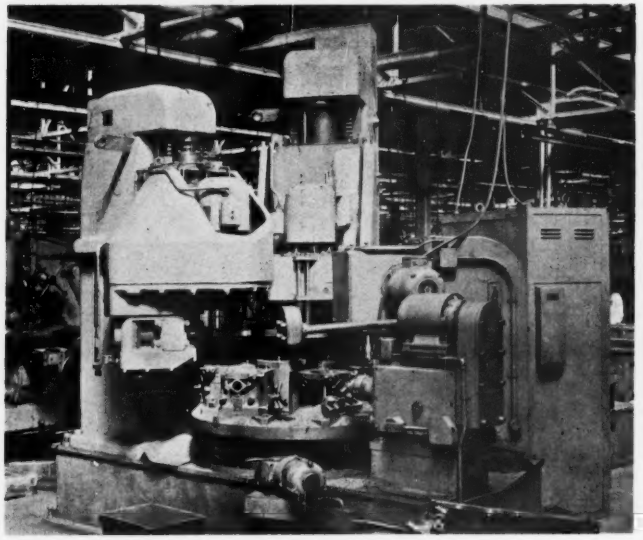


Fig. 58. Special-purpose Archdale machine for drilling, tapping and milling the water pump body

Piston grinding is the last operation for which the centre in the crown is needed. Therefore, the pip is removed at the next operation which is carried out on an Adcock and Shipley hand milling machine. Two slitting saws are mounted on the arbor of this machine. One removes the pip from the crown while the other is used to saw two heat insulating slots in the scraper ring groove. Following this, eight oil holes are drilled in the scraper ring groove and four are drilled in the gudgeon pin bosses. A special Pollard machine, illustrated in Fig. 56, is used. It has eight horizontal spindles which are radially adjustable. They are simultaneously controlled by one common cam feed which is engaged by movement of a pedal. For drilling the holes in the gudgeon pin bosses there is a centrally-mounted two-spindle vertical standard drilling head. This head is fed by hand. The cycle time per piston is only eight seconds.

At this stage the piston is tin plated in a Canning automatic plant. After being tinned, pistons are mounted two at a time on a two-spindle single-end Precimax fine borer, see Fig. 57, for finish machining the gudgeon pin holes. Solid tungsten carbide boring bars, each carrying two solid tungsten carbide tools are used. One tool on each bar semi-finishes and the other finishes the bore. To ensure correct mounting, there are two alignment bars mounted on the workhead of the machine. Each is accurately aligned with one of the machine spindles. They are advanced independently through the gudgeon pin holes to ensure that the axis of the part finished holes is in alignment with the spindle axis before the clamps are applied. To complete the machining, a slot is cut in the skirt from the bottom to the scraper ring groove. A hand-operated Adcock and Shipley milling machine is used for this operation. After machining is completed, every piston is given a complete inspection and is graded

Fig. 59. Archdale special-purpose drilling and tapping machine with an annular indexing table

for weight before being transferred to the assembly section.

Water pump body machining

One of the most outstanding of the many interesting production developments in connection with the new engines is the machine line for the water pump body. For this component James Archdale and Co., Ltd., have once again co-operated with the engineers of Vauxhall Motors Ltd. to develop a line which will minimize the amount of handling required. Five machines in all suffice to complete the machining, which is of a fairly complex character. The machines are, an Archdale miller, three Archdale multi-station multi-spindle machines and a Precimax double-end fine borer. In comparison with the methods previously used, this line has reduced the machining time from something in the order of 28 minutes to approximately nine minutes. At the same time it has also reduced the number of machine operators necessary.

The water pump body is of cast iron. At the first operation the joint face is milled on an Archdale vertical miller. After the milling operation the casting is transferred to a special-purpose Archdale drilling, milling and tapping machine with a 48 in diameter six-station indexing table. Although this is a special-purpose machine, all the heads are standard Archdale units.

Loading and unloading are effected at the first station. The work-holding fixtures incorporate equalizing jaws to give correct location. At the second station there is a standard vertical hydraulic feed unit with combined milling and multi-drilling attachments. The drilling head is vertical. It is tooled for drilling eight holes in the joint face and for core drilling a $\frac{7}{8}$ in diameter central hole. The milling head is horizontal and carries a 5 in diameter, eight-teeth Galtona cutter for milling the flange of the water outlet.

A vertical, hydraulic feed, fixed centre type boring head serves stations three and four. At station three there is also a horizontal cam-feed drilling unit. While the component is at the third station, the vertical head is used for drilling three holes and countersinking two holes in the joint face, for turning the outside diameter of a central boss and for boring a $2\frac{1}{16}$ in central hole. At the same time the horizontal head at the third station drills two holes in the water outlet flange and a hole through the body to the centre bearing hole.

When the table is indexed to bring the component to the fourth station, the remaining tools in the vertical head common to the third and fourth stations come into action. At this station two of the drilled holes in the joint face are reamed to give accurate locations for subsequent operations and four tapping holes in the joint face are countersunk. In addition, a combination tool semi-finishes two faces to leave an allowance for fine facing at a later operation.

A vertical and a horizontal tapping head are arranged to work simultaneously at the fifth station. These are standard Archdale tapping heads, incorporating independent lead screws to allow independent tap setting and telescopic collets to ensure freedom from tap breakage should any of the holes be incorrectly drilled. The vertical head is used for tapping five holes in the joint face, and the horizontal head for tapping two holes in the water outlet flange.

At the sixth and final station there is a horizontal cam-feed unit with a single spindle. It carries a combination tool for counterboring two diameters in the water outlet. This machine, which is shown in Fig. 58, has a power-operated table and an automatic cycle. The cycle is initiated by push button and is: table indexes; heads feed down or in; heads retract and spindles stop. Unloading and loading can be effected during the working cycle. Incidentally, the two machines immediately following in the sequence also have power-operated indexing tables and similar cycles.

For the next operation an Archdale special-purpose drilling and tapping

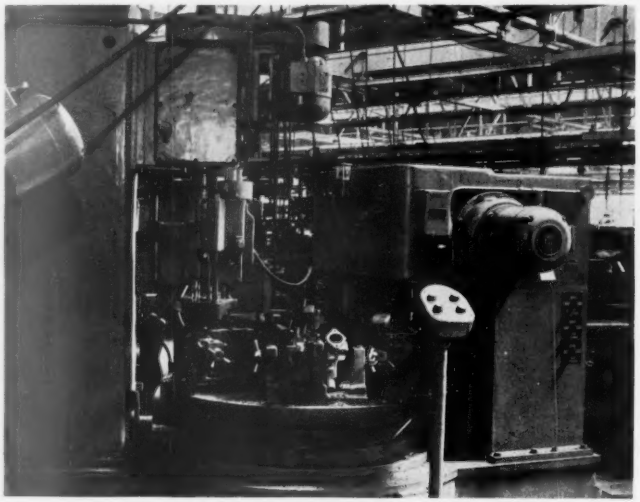


Fig. 60. Archdale drilling and tapping machine with five-station indexing table. Each component makes two cycles on the machine

machine is used. This machine is illustrated in Fig. 59. It has an annular five-station indexing table. At the first working station there is a single spindle vertical head with hydraulic feed. It carries a combination tool for core drilling two central holes.

Three heads are arranged to operate at the third station. One is vertical and the other two angular. The vertical head has hydraulic feed and is tooled to form a recess in the central bearing bore. One of the angular heads is placed outside the table while the other is mounted on a base that passes through the annular indexing table. These are cam feed units, and each head carries a single drill for drilling a hole through the boss of the casting into the bore for the bearing. At this station all three heads do not work simultaneously. On the contrary, the vertical head feeds down to form the recess in the bearing, while the angular heads remain stationary. In-feed of the angular heads does not start until the vertical head has started to retract. If these three heads were not so synchronized, the tools in the angular heads would foul the tool in the vertical head. At the third station the central boss and five holes in the joint face are spotfaced from a vertical head, while at the fourth station one hole is tapped from a vertical head and one hole is tapped from a horizontal head at the fifth station.

Another special-purpose Archdale drilling and tapping machine, see Fig. 60, is used for the next operations. It has a five-station indexing table with two work-holding fixtures at each station. Of each pair of work-holding fixtures, one is designed to hold the component in one position while the second holds it in a completely different position. Therefore, each component has to go through two cycles, one in the first and one in the second work-holding fixture.

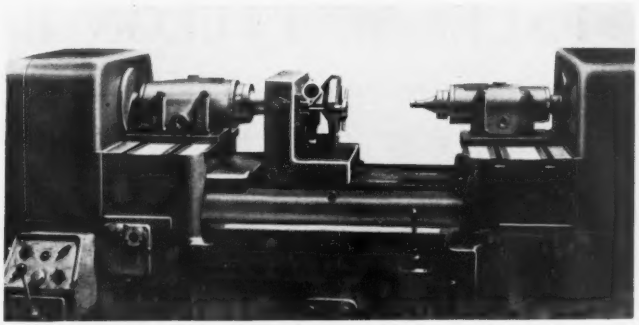


Fig. 61. The set-up for fine boring and fine facing the water pump body on a Precimax fine borer

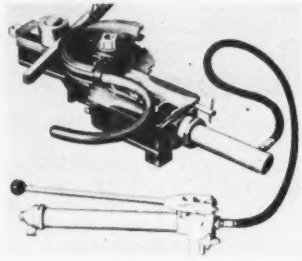
At the first loading, and the first station, a tapping hole is drilled and another hole is drilled and counterbored with a combination tool. From the same vertical head, but at the second loading, two tapping holes are drilled. There is a single-spindle vertical head at the second station. This drills a hole in the component at the first loading, but is an idle station for the second loading. A common tapping head serves the third and fourth stations. At each of these stations and at each loading, one hole is tapped. To complete the machining, the component is transferred to a double-end Precimax fine borer, see Fig. 61, which is tooled for fine boring the bearing bore and fine facing two seats. Incidentally one face must be square with the bearing bore within 0·0008 in total indicator reading.

The examples quoted are far from exhausting the interesting developments for machining Velox and Wyvern machine details, but they do illustrate the manner in which complex production problems have been solved by close co-operation between the design and production engineering departments at Vauxhall Motors Ltd., followed by equally close co-operation between the production engineers and the machine tool manufacturers. It is worth remarking that almost without exception the machines used in the production of the engine details are of British design and manufacture. That the machines installed are suitable for producing accurate work at high rates of output is only to be expected; what is really noteworthy is the manner in which wherever possible individual heads have been grouped to form one machine, thus minimizing the amount of handling required without incurring the complexity of multi-station transfer machines.

EXHAUST PIPE BENDING MACHINE

DESIGNED specifically for bending exhaust piping and other light gauge steel tubing, this new machine is virtually a lightweight version of the standard hydraulic pipe bending equipment manufactured by Chamberlain Industries Ltd., Staffa Works, Leyton, London, E.10. The work is well supported by an accurately machined centre former and a back former of the knife-edge type which fits in the overlapping flanges of the centre former and is firmly positioned by a roller on the bending arm. A ratchet system transforms the straight-line thrust of the hydraulic ram into the rotary action necessary for bending thin-walled tubing.

Bends may commence one inch from the end of a tube and can be continued up to a maximum of 180 deg. One full stroke of the arm produces a bend of 60 deg, and at the end of the stroke a pawl holds the ratchet to counteract any tendency of the tube to spring back. An angle indicator is provided so that the degree of bend achieved can be readily observed during operation. Bends can be made in various planes, and directly connecting 90 deg and 180 deg bends obtained without waste of tube.



The Staffa pipe bender

The machine can handle seamless tubing of 16 to 19 gauge in outside diameters of 1 in to 2¼ in. Formers are available for this range of diameters in steps of ⅛ in. (2022)

Cutting Fluids

A NEW edition of the Fletcher Miller publication "Cutting Fluids" is now available. Enlarged in scope and well illustrated, it deals comprehensively with the selection and use of cutting fluids for various machining operations on a range of different materials. A chart of recommendations indicates appropriate fluids for turning, milling, grinding, screwing and tapping in ferrous and non-ferrous metals and alloys. Copies can be obtained on application to the manufacturers, Fletcher Miller Ltd., Alma Mills, Hyde, Nr. Manchester. (2021)

FACEL-MÉTALLON BODYWORK

French Methods for Coachwork in Light Alloy, Steel, or Combinations of the Two Materials

By A. G. Douglas Clease, B.Sc., A.M.I.Mech.E.

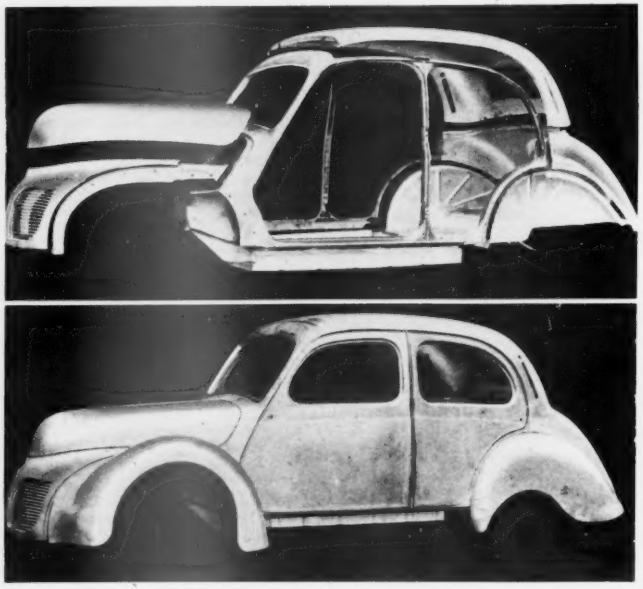
FOUR body exhibits, of different character but made by the same firm, commanded attention at the Paris Salon last year. These had two features in common, they were of all-metal construction and each was outstanding in its class for its lines and well-balanced appearance. The productions of Companie Facel-Métallon, they were respectively :

(1) The Dyna-Panhard four-door saloon
(2) The Simca-Sport coupé
(3) The Ford Comète two-door, four- five-seater saloon
(4) The 4½-litre Bentley two- three-seater coupé.

It is of interest to note that Facel-Métallon are not coachbuilders in the accepted sense of the word but specialists in metal working. Thus, in addition to the building of all-metal bodies in series, they also manufacture bumpers, nave plates, wheel embellishers and radiator grille frames in stainless steel for the automobile industry generally. For the aircraft industry they produce combustion chambers for Rolls-Royce Nene and Tay and for De Havilland Goblin turbojet units. Other activities are the manufacture of pressed steel kitchen cabinets, stainless steel sinks, and of a range of pressed steel furniture for office use.

Production of vehicle bodies of various types reaches an average total of 105 per day. This is expected to be increased during the early part of this year to 125 per day. The present production figure is constituted approximately of 70-72 Dyna-Panhard four-door, four-light saloons, between 12 and 15 Simca-Sport coupés and Ford Comète two-door saloons and from 20-25 bodies for the V.L.R. (voiture légère routier), a military vehicle somewhat similar to those being produced in this country.

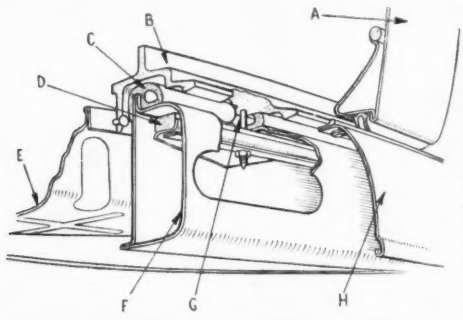
Four works are engaged in the manufacture of these various products. Possibly the most important since its products are the basis for the work of the other factories, the press shop is the largest and is situated at Amboise in the Loire Valley about 130 miles south-west of Paris. Another is at Dreux, some 50 miles west of Paris, and the other two are at Colombes and Courbevoie in the suburbs of the capital. The steel and aluminium pressings from Amboise are assembled into the body shells at Colombes. They then go to Dreux for trimming and painting. The Courbevoie plant is small and has been used as a pilot plant for the production of prototypes and body jigs, but it is now a research and development centre.



Ten Alpax castings form the Dyna-Panhard body frame

At the present time the Amboise plant is strained for sufficient covered floor space, although there is ample room to expand. It houses a Lake Erie double-action, 575-ton press employed for the larger pressings such as roof panels, a Clearing 500-ton, triple-action press and a Spiertz double-action, 180-ton press. There are in addition various small presses used for the lesser components. In all some 3,000 different pressings are produced for the various items manufactured by Facel-Métallon. Three additional presses are shortly to be installed. These are a Clearing 750-ton and two Spiertz, one a 450-ton and the other a 320-ton. When installed these should materially increase the output of the plant which is at present working three shifts a day.

It is at Amboise also that the stainless steel components and the kitchen cabinets are produced. For the bumpers, of which between 400 and 500 pairs per day are produced for the Facel bodies and also for other car manufacturers, an ingenious automatic polishing line has been designed and installed by the plant engineers. It is some 30 yd in length, and a chain conveyor carries the bumper pressings through a line of sixteen polishing stations. The polishing mops are individually driven by electric motors and are mounted on pivoted beams to bear with the desired pressure on the bumpers. They are also inclined to the line of work at different angles, so that the whole surface of the bumper section is effectively dealt with.



A Door
B Body sill casting
C Rubber bearer
D Rubber support
E Floor
F Side frame
G Set pin
H Side valance

Mounting of body on chassis frame, Dyna-Panhard

The conveyor is loaded at one end with unpolished pressings and the finished bumpers are removed at the other end. A very high degree of polish is obtained on the bumpers, and also on other stainless steel components such as the nave plates, radiator grille frames and wheel embellishers, which are handled individually on conventional polishing equipment.

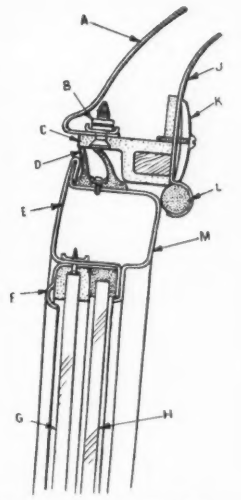
At Colombes the Dyna-Panhard bodies are assembled. These four-door, four-light saloons are of light alloy construction and possess several unorthodox features. The scuttle consists of two castings in Alpax, the composition of which is iron 0·75 per cent maximum, silicon 12·2–13·0 per cent, impurities 0·2 per cent maximum, remainder aluminium. The lower part of the scuttle is formed by one casting and the upper part, together with the screen pillars and roof peak, by the second casting. Other Alpax castings form the sills, standing pillars, cant rails and front half of the rear wheel arches. These ten items are all sand castings, not die-castings as might be expected. The reason is that when production began in 1945 there was little experience in the production of large die-castings. It has since been found that where large castings are concerned the rate of production is not much higher when using metal dies and hardly justifies the extra cost of the dies. These sand castings are, however, clean and require little work on them other than the trimming, with portable abrasive disc equipment, of exposed surfaces. The various abutting surfaces are machined, drilled and tapped to receive the bolts and screws which are used to assemble the structure.

When the castings have been machined they are assembled in jigs and bolted together so that they form the frame work of the body. The only steel pressings are three which are spot-welded together to form the front portion of the floor. Ribs are formed in them in the pressing operation to act as stiffeners. The floor is then attached to the sill castings by riveting, a sealing strip being interposed in the joints.

All other pressings are in Duralinox, the composition of which is copper 0·1 per cent maximum, magnesium 3·0 per cent, iron 0·5 per cent maximum, silicon 0·25 per cent maximum, manganese 0·2–0·5 per cent, remainder aluminium. These pressings are mostly produced from 0·047 in sheet, although for the bonnet pressings 0·039 in sheet is used.

The rear floor and wheel box is a sub-assembly built up of such pressings spot-welded together. The wheel boxes consist of a flanged rim and double side plates having radial swages on opposing sides, so that these form tubular stiffeners when the plates are spot-welded together and to the flange of the rim. This sub-assembly is bolted to the wheel arch castings and is supported at the rear on the wheel suspension brackets.

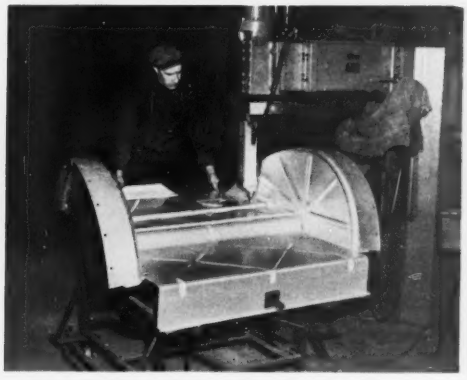
Two large pressings form the body shell. One of these is the boot panel which finishes at waist level and has a circular depression to receive the spare wheel. Into this depression is spot-welded a separate dished pressing which locates the wheel. The other large pressing is the roof panel, in which are pierced the rear window



A Roof panel
B Self-locking nut
C Cast cant rail
D Rubber seal
E Door outer panel
F Window framing
G Sliding light
H Fixed light
J Head lining
K Trim moulding
L Rubber beading
M Door inner panel

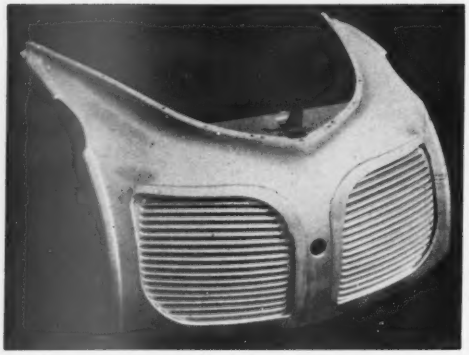
Section through door and cant rail

Automatic polishing line for stainless steel bumpers



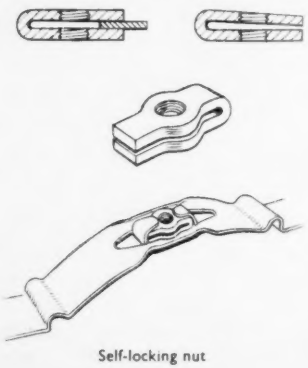
Spot-welding the rear floor and wheel boxes

aperture and apertures for the direction indicators. The front edge is flanged and is bolted to the roof peak with a sealing strip interposed. The side edges of the roof pressing are also flanged and carry self-locking nuts located on flexible strips attached to the pressing. Countersunk screws pass through the cant rail castings into these nuts, which are made from steel strip bent into a flat U shape and reset after tapping so that they give spring pressure on the threads. The bottom edge of the roof pressing overlaps the top edge of the boot pressing and is spot-welded to it, a polished beading being applied to hide the joint.



Dyna-Panhard front cowling and grille

Doors consist of two pressings, the outer panel of 0·039 in material and the inner stiffener of 0·047 in thickness. The outer panel is flanged to form the window aperture and the inner panel is also flanged to form the door edges, the two pressings being spot-welded together and the edge of the outer panel also being turned over the edge of the inner pressing. Reinforcements are added for the hinges in the form of angle brackets which are fixed to the inner pressing. A channel-section reinforcing strip also extends the whole length of the hinge edge of the door, and another channel section provides for attachment of the lock and handles. Round the large aperture in the centre of the inner pressing runs a rubber bead to prevent drumming.

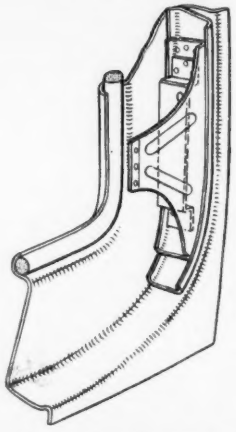


Self-locking nut

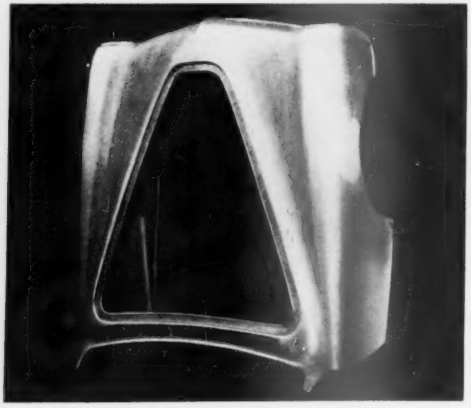
Forward of the dash, and bolted to it, extend two pressings which carry the front assembly of bonnet and front cowl. The cowl is pierced for the simulated radiator grille, which is a separate louvred pressing spot-welded into position. As the engine is air-cooled the louvres admit air direct to the cylinders and not actually to a radiator. The bonnet top consists of two pressings spot-welded together down the longitudinal axis and reinforced by a transverse stiffener at the back edge and by a triangular plate spot-welded into the flanged nose of the bonnet top at the front end. Use is made of this stiffener plate to carry the bonnet securing catch.

Wings are separate pressings in Duralinox and are bolted to the main structure so that they are easily removed and replaced in the event of damage. Valances finish off the bottom sides of the body and are attached by screws to the alloy sill castings. The bottom edge of the valance is wired for stiffness so as to prevent drumming.

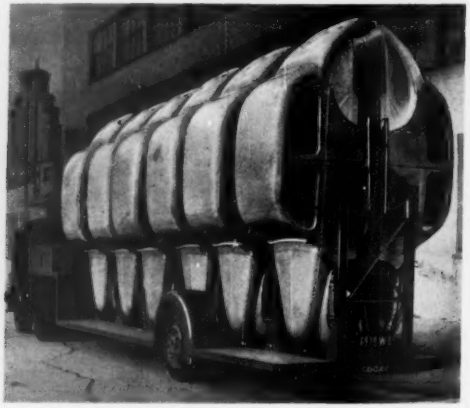
Assembly takes place on wheeled stands which are moved by hand along a guide rail. Owing to the



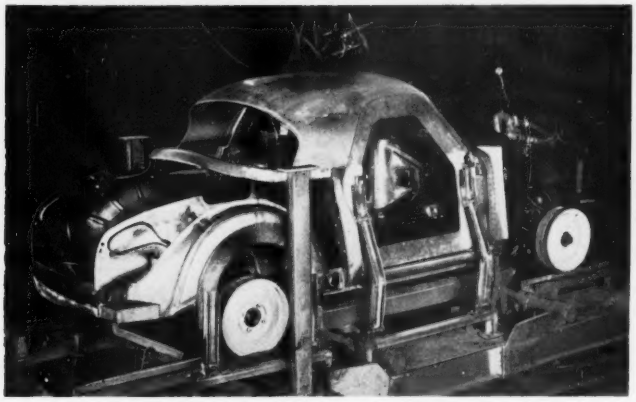
Detail of door construction showing hinge reinforcement and rubber beading on inner panel



Simca-Sport bonnet and wing assembly



Twelve Dyna-Panhard body shells are transported on a semi-trailer



Main assembly jig, Simca-Sport body

lack of space the bodies follow one another very closely at times, leading to what would appear to be somewhat cramped working conditions. This, however, does not seem to worry the operatives or to interfere with their normally fast speed of working. The time taken for the production of a Dyna-Panhard body shell, unpainted and untrimmed, is 30 man-hours. The bodies are mounted, trimmed and finished by the Panhard company in Paris.

Stainless steel facia on Ford Comète

For the transport of the body shells a Coder low-loading semi-trailer is used. Twelve shells, with light tubular frames attached, are mounted vertically in pairs and secured to seven vertical supports on the trailer by toggle-action clamps. As the weight of a shell with its tubular support frame is only about 300 lb, the total load is approximately 33 cwt.

In the coupé bodies for the Simca-Sport chassis the system of construction is different. Steel pressings suitably ribbed for stiffness are used for the floor, and a framework of pressed or folded steel members is built upon the chassis with the use of jigs to position the various units during welding operations. The sills, scuttle dash and door hinge pillars, door shut pillars, and rear waist rail are constructed in this manner. The inner part of the rear wheel arches is also a pressing which extends back to receive and support the boot or tail panels. Projections from these pressings also connect them to the waist rail, to which they are seam welded, so that the waist rail is, therefore, supported by them.

Considerable use is made of spot-welding, but oxy-acetylene welding is employed for certain seam joints. On this framework is mounted and welded the steel roof pressing, which ties both scuttle and rear portions together.

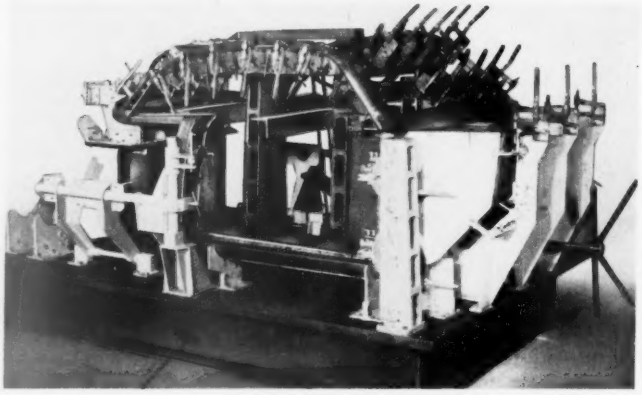
Two distinct panel units, or rather assemblies of panels, are the front wings, bonnet surround and cowl at the front and the boot, rear wings and sides at the rear. The front portion consists of four main pressings in aluminium seam-welded together. For the rear portion Duralinox pressings are used, again there being four main pressings welded into a whole. Round the boot lid and bonnet lid apertures channel sections are pressed, and these serve as drain

Simca-Sport body assembly line

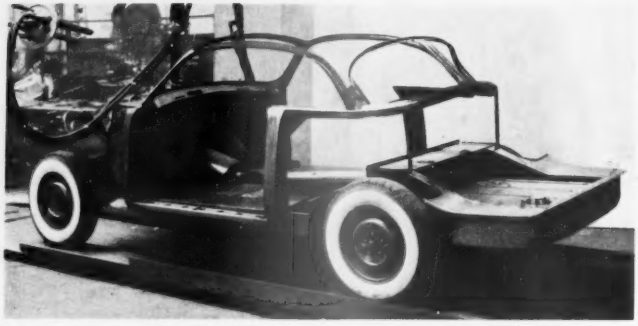
gutters and also stiffen the pressings. For the same reasons the edges of the wheel apertures in the wings are flanged.

The joints between the steel roof pressing and the aluminium front panel assembly, and between the roof and the Duralinox boot panelling, are made by a process described as "cold welding". A special tin-bearing alloy is used, joint surfaces being heated by ordinary gas torch, and the alloy on melting combines with both the steel and aluminium or light alloy.

Pressings for the Ford Comète body are all of steel, but the system of construction is similar to that followed for the Simca-Sport coupés, although there is an important difference in that the body frame is integrated to the chassis by welding. First, however, a modification is carried out on the chassis. The upper half of the rear portion of the cruciform cross-bracing is cut away by oxy-acetylene torch, reducing its depth by half, and suitably reinforced so that there is no loss of strength or torsional rigidity. This modification, approved by the Ford engineers in France, is to lower the floor level at the rear.



Ford Comète main structure jig



A steel body structure is welded to the Comète chassis

Box-section sills and pillars, formed by pressing and folding, constitute the main part of the body frame. Other pressings form the scuttle bulkhead and the rear waist rail. The last-mentioned extends round the back of the car and unites the two door-shut pillars, which finish at waist level. At the back two vertical supports connect the waist rail to the main chassis members.

Other channel-section pressings forming the cant rails and the screen rail, are welded to the rear waist rail, to which is also welded another channel-section pressing supporting the rear end of the roof pressing and conforming to the shape of the rear window.

To this structure the roof pressing is welded. The bonnet surround and

This low-built, special Bentley weighs 3 cwt. less than a standard saloon

front wings form a sub-assembly of panelling at the front, while the rear wings and boot likewise form a unit of panelling at the rear. Much use is made of spot-welding, and the joint between the roof panel and the boot pressing is so made. It is not finished off by loading with solder but is disguised by a bright beading.

Doors are built up of exterior and interior panels, so that in effect they are very rigid box sections in themselves. Reinforcements are added at the points where the concealed hinges are attached, and also for the locks and handles. The doors finish at the waist rail, and polished metal frames are attached to form the glass runs; metal frames also carry the fixed quarter-lights.

Steel pressings also serve as the foundation of the seats, both for the cushions and squabs, and in the case of the rear seat they add, therefore, to the general stiffness of the body shell. A stainless steel pressing is used for the facia, and an interesting point is that the surface is finished and polished before the pressing is made.

A point of note about the Bentley coupé is that it is a "one-off" production. It has been built for the chairman and managing director, M. Jean Daninos, who combines also the duties of designer and development engineer. The system of construction followed is similar to that used for the Ford Comète, in that a steel side, front and rear structure is built up in the form of a frame to which the panelling is applied. This body frame structure is attached to the chassis at four points by welding.

The panel pressings are all of Duralinox, with the exception of the doors which are built up from steel pressings. Another interesting feature is the burr walnut finish of the pressing which forms the main part of the facia and swells out in front of the driver to carry the instrument panel. The finish exactly matches that of the real timber of the panel in front of the passenger and the garnish rails. Use is made of the deep section of the steel doors to receive compartments containing toilet articles and flasks, access to which is given by hinged or sliding panels in the door trim.

When prototypes are being developed, or on other occasions when it is expedient to do so, use is made of Kirksite dies and a drop-hammer to produce the panels. The zinc base dies are made by the plant engineers.

Much use is made of spot-welding, both for light alloy and steel. Before welding, panels are treated with a preparation known as Framanol, consisting of phosphates of chromium, triethondamine and alcohol, which degreases and dissolves the surface oxide film. Argon arc welding has been under investigation and is likely to be adopted for certain joints in panels where smooth weld beads are essential for a high finish.

Painting and trimming are carried out at the Dreux works, where infrared drying ovens are installed. Methods here are orthodox and the finish obtained is of a high standard. Much use is made of Dunlopillo for cushions and squabs, and a popular upholstery fabric for the Simca-Sport coupé is corduroy. Plain cloth is largely used for the Comète saloons and the Bentley coupé is trimmed in Connolly leather. Seats are carefully designed on anatomical principles, to provide adequate support where it is needed.

From the foregoing details it will be realized that somewhat novel methods of body engineering have been developed by Facel-Métallon. It is also a point of interest that the system evolved appears to lend itself equally well to production in series or to the fabrication of a "one-off" body.

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

Material Handling in Works Stores

By L. J. Hoefkens.

London: Published for *Mechanical Handling* by ILIFFE AND SONS LTD., Dorset House, Stamford Street, S.E.1. 1952. 5½ × 8¼. 106 pp. Price 10s. 6d.

The fact that a study of the possibilities of mechanical handling improvements in any industry is worth while is made abundantly clear in this book, written by the Stores Controller of a large group of engineering companies. The basis of the work is an account of the introduction of improved techniques in a typical plant. The progress of material from its raw state to the final despatch of finished goods is traced by means of practical descriptions in straightforward terms and a large number of informative illustrations.

The outstanding piece of equipment used in this connection is the fork truck with pallets, and a large section is devoted to this subject, with details of different types and their uses, and also the various attachments which are available for special purposes. An early section of the book gives assistance on the initial planning necessary before adopting new methods, detailing point-by-point the basic principles to be considered. The newcomer to a completely mechanized handling system is thus forewarned of pitfalls which otherwise might be encountered. The rest of the book is devoted to a description of a system in actual practice, with useful suggestions on a variety of points such as identification, numbering and material movement control. There is also a section dealing with the handling of swarf and scrap, the mechanization of which is just as advantageous as in the case of raw materials and finished products.

The author points out that apart from economy in labour and in working and storage space, a properly organized handling system is valuable in other directions, such as in the simplification of planning incentive schemes, or in the reduction of many of the worries of stock-taking. In his conclusions, the author reviews the results of the introduction of the new system in the works described, and that it was a successful enterprise is illustrated by the fact that the present cost of receiving, storing and despatching a single unit of production is approximately half that of the previous cost. Other points to be noted are that the number of handling stages is greatly reduced, thus eliminating risk of damage; increased output has not necessitated additional storage space or new buildings; movement of material has been speeded up at all stages; stores paperwork systems have been simplified, thus reducing the risk of inaccuracies; less labour is needed to handle increased output; and any future movement or reorganization of stores will be simple and inexpensive.

This country is still backward in making use of all the latest equipment and devices that she produces, but Mr. Hoefkens' book indicates the way to remedy this state of affairs. The results of years of experience, careful operational study, and practical development are brought together in a work which will smooth the way for any management considering the adoption of a new system and greatly assist in the reorganization of any system already in existence but not fully efficient.

Tractors and Their Power Units

By E. L. Barger, W. M. Carleton, E. G. McKibben and Roy Bainer.

New York: JOHN WILEY & SONS, INC. Obtainable from Chapman & Hall Ltd., 37, Essex St., London, W.C.2. 1952. 5½ × 8¼. 496 pp. Price 52s.

This is stated to be the first book (in the U.S.A.) on tractors written on a technical engineering level by top men in the field. Let us first see who these men are. They are, respectively, the Manager of Farm Practices Research of Harry Ferguson Inc., Associate Professor of Agricultural Engineering at Michigan State College, Principal Agricultural Engineer at Usda Tillage Machinery Laboratory, and Professor and Chairman of the Division of Agricultural Engineering at California University. The pooled knowledge of these authors, together with the information that they have extracted from published papers over a number of years (all of which are referenced for the reader's further study) provides an up-to-date survey of the farm tractor as developed in the U.S.A.

Well-illustrated, simply written and

profusely sub-headed into short sections, this is a book for all students of agricultural engineering as well as their teachers. It is also a book for all who concern themselves with the design or regular use of farm tractors. It deals not only with the engine, but also with the operator's comfort and convenience, the mechanics of the chassis, hitches, clutches and brakes, power take-off, hydraulic controls, traction devices and tractor power cost estimating. Its more technical sections cover power and power measurement, thermodynamic principles, adiabatic changes, gyroscopic couple, combustion and heat values. These sections, however, can be omitted by the less technical reader, although they are very necessary to the ambitious student. Certainly, the mathematically-minded agricultural engineering student will gain most from this book and it is he, and tractor designers, for whom the book is almost a necessity.

The fact that it is written about American tractors and practice will not detract, it is thought, from its value in this country. Its only real handicap is its price, which the dollar rate of exchange makes so high. Even so, it still provides value for there is information in this volume that would be difficult to gather elsewhere. Only the information given in the Appendix, which quotes a number of S.A.E. specifications, and gives many American sizes for tyres, etc., are of little value outside America, but the A.S.M.E.-S.A.E. Tractor Testing Code is of interest.

A criticism might be levelled at this book on the score that a considerable amount of the information given is applicable to other machines, especially to internal combustion engines, and that its value is no different when applied to tractors. This might be so in many cases, but certainly not in all, and in any case, the agricultural engineering student does not always have the opportunity of studying many other books on related subjects. It is more convenient and time saving to have the most valuable parts of this data to hand in one volume. In this book, only four chapters deal with conventional material on internal combustion. The authors claim that the other twenty-two chapters present new material now found in book form for the first time.

Typical problems follow each chapter, but many of these are numerical ones and lose their significance to the student when no answers are provided. The value of a technical book of this type is enhanced if the student is able to work problems and to know whether he has solved them correctly. This is an omission that should certainly be rectified in a subsequent edition.

Productivity and Probability

By T. F. O'Connor, B.Eng., A.M.I.Mech.E.

Manchester: EMMOTT & CO., LTD., 31, King Street West. 1952. $4\frac{1}{4} \times 7$. 193 pp. Price 5s.

The author has taken it upon himself to prepare a better book on the subject of time-study, productivity, etc., than has been written before—he tells us so in his preface. He states that present books deal too much with facts that anybody who is not a cretin could discover for himself after a week in a factory with a stop-watch. This may be true in many cases but it might well be that readers do not have the opportunity of spending a week in a factory with a stop-watch and would much rather buy a book which tells them all about it. The author does state, however, that there are one or two exceptions but it became clear to him some time ago that if he wanted a good text-book on the subject he would have to write it himself.

In spite of these opinions, the author has provided a valuable monograph (this book, incidentally, is No. 65 in the Mechanical World series of Monographs). The book is also enriched by the inclusion of some of the Ashcroft Tables and Numbers with regard to the references to machine interference.

The practical time-study engineer must find much of value in the book because the author's object is to show how industrial efficiency may be improved. Therefore, the book should find readers amongst many factory executives. Details concerning the training of time-study men or how a time-study department should be run are not included, for surely this information can be found in any book on the subject. The author deals with more fundamental and practical points than these and endeavours, not unsuccessfully, to give the answers to practical problems that arise when production has to be increased and a ready means of solving this problem has to be found at once. The "get-down-to-business" and practical approaches of the author are evident throughout the book. Details of the advantages and working of bonus schemes are included and these go into details about work sheets, standard relaxation, non-piecework schemes, the Rowan Plan, etc. Following this, and in correct sequence, Machine Work is discussed.

Throughout the book, the problems of human relationships are sympathetically appreciated. Possibly the author's reflections on this subject are worth quoting, for there is a great deal of truth in them. He says: "In itself it is neither good nor bad but neutral. It is a tool of management which can be used for beneficial purposes or the reverse. If the creed of philosophy is wrong, if we believe that a section of the community is there for exploitation, the result cannot be but evil. If on the other hand we believe that the community is one and indivisible and that the heritage of the past is something on which society depends and in which all have a share of ownership as co-heirs of the past, provided only that they contribute each according to his talent or ability, then there is the probability that we may do some good, though good intentions are not enough."

These remarks might introduce many subjects that involve the factory worker who too frequently believes that he is being exploited to provide increased productivity. Therefore it is good to see that the author of this book is fully aware of the problems of the time-study man.

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The Practical Engineer Pocket Book

Edited by N. P. W. Moore, B.Sc., A.C.G.I., D.I.C., A.M.Inst.F.

London: SIR ISAAC PITMAN & SONS LTD., Parker Street, Kingsway, W.C.2. 1952. $3\frac{1}{2} \times 5\frac{1}{2}$. 744 pp. Price 12s. 6d.

The 64th edition of this pocket book includes certain revisions, especially in the sections Machining of Metals and Welding and Cutting, these being mostly re-written. The Steam section now includes notes on the solutions of problems in steam engineering by the use of steam diagrams. Many half-tone illustrations have been replaced in various sections but some of the existing ones also need renewing for two reason. (1) They are not sufficiently distinct to be of value to the reader and (2) there is insufficient variety, in some cases, amongst the manufacturers' goods that they illustrate. More companies should be contacted to lend illustrations of their products.

The book retains its character and deals with subjects under such headings as Cranes and Lifting Tackle, Locomotive Practice, Steam Turbines, Gas and Oil Engines, Gas Turbines, Air and Ventilation, Hydraulics, Lighting for Workshops and Offices. It is full of detailed information and includes charts, graphs, tables, and all the references that one would expect to find in an engineer's pocket book.

Criticism of what has been included or what has been omitted is easy in any book of this type. Here and there, it might well be thought that too much specialized information is included at the expense of more general educational notes. For example, the section on Lubrication deals mostly with lubrication by graphite—an important subject but only one small aspect of the great lubricating field. It would appear that some sections have been written by the too specialized writer.

Undoubtedly the book will continue its popularity since it includes a vast amount of technical detail that is commonly needed for daily reference by all practical engineers, and also, we imagine, by many theoretical ones too. The German-French- and Spanish-English technical dictionaries are again included and are useful for those corresponding overseas on technical matters.

NOISE MEASURING INSTRUMENTS

Some Installations and Methods

By W. Bye, B.Sc. (Eng.)

THERE is no convenient way to express, in a manner easily understood, the degree of quietness of a car. Nevertheless, a noisy gear or a metallic rattle in a motor body can play havoc with sales and may well, in the purchaser's mind, offset many of the good points of the vehicle. This is particularly true in the family car class, as women, who usually are more sensitive to unpleasant noises than men, exercise a marked influence in the selection of these cars.

It is not surprising, therefore, that many leading manufacturers take considerable care in checking the sound and vibration levels in their cars. Mention has already been made of the difficulty in defining the magnitude of a noise. This difficulty is, to a great extent, due to the different effects of identical sounds on different observers, as well as to the fact that the sensitivity of an observer varies with changes of frequency and sound level. Clearly, an instrument must measure the loudness of a sound by means of its objective characteristics such as energy per unit area, or pressure level. Therefore, the requirement for a useful measurement is to relate this objective characteristic to the actual subjective response on the human ear.

How this is done in a commercial sound level meter can be followed from a study of Fig. 1, which shows the Fletcher-Munson curves for equal loudness to the human ear. The ear is much more sensitive to soft noises than to loud ones. This is one reason why the level of sound is expressed in terms of the logarithm of the ratio of the energy of the sound to the energy which can just be discerned by an average human ear. Zero decibels is defined as the lowest sound that can be heard, while at a level of 120 decibels the sensation of hearing changes to one of pain. Between these two limits, the range represents a multiplication of the sound energy by a factor of no less than a billion, or 10^{12}. It is this tremendous range which, if at all possible, should be accommodated by a single instrument. This makes difficult the design of instruments for sound measurement and is a further reason why logarithmic units of measurement must be used to compress it into a scale of convenient size. A quantitive indication of the energy of sound may be derived from the fact that a sound energy of only 2,000 watts is all that would be necessary to maintain a noise of medium loudness over the whole of Wales.

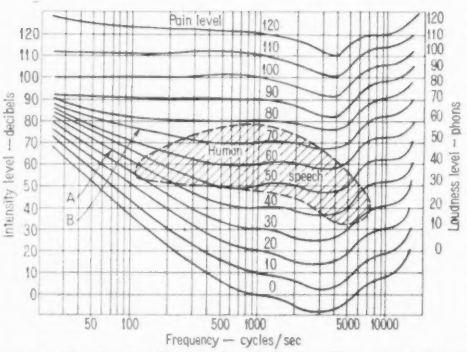


Fig. 1. The Fletcher-Munson "equal loudness" curves showing the relationship between the actual frequency and loudness of sounds, and their apparent loudness as perceived by the human ear

Returning to the curves, it will be seen that the sensitivity of the human ear falls off rapidly at high sound levels, and at fairly low as well as very high frequencies. There is, in fact, the possibility of sound reduction by moving the frequency of the noise into the region where the ear is least sensitive. This variation of sensitivity with both sound level and frequency renders it impracticable to manufacture an instrument with exactly the same characteristics as the ear. What has to be done, therefore, is to include a number of weighting networks which have, at a particular noise level, the same frequency response as the ear.

In the instrument illustrated, the noise levels chosen are the 40 phon and the 70 phon equal loudness contours marked A and B on the Fletcher-Munson Curves,

THE DECIBEL RATING OF COMMON SOUNDS

	Decibels		Intensity units	Sound pressure Dynes/cm²
DEAFENING	120	Threshold of pain	1,000,000,000,000	200
	110	Thunder: artillery Nearby riveter Underground train Boiler factory	100,000,000,000	
	100		10,000,000,000	20
VERY LOUD	90	Loud street noise Noisy factory Lorry, unmuffled Police whistle	1,000,000,000	
	80		100,000,000	2
LOUD	70	Noisy office Average street noise Average radio set Average factory	10,000,000	
	60		1,000,000	0·2
MODERATE	50	Noisy home Average office Average conversation Quiet radio set	100,000	
	40		10,000	0·02
FAINT	30	Quiet home Private office Average auditorium Quiet conversation	1,000	
	20		100	0·002
VERY FAINT	10	Rustle of leaves Whisper Sound-proof room Threshold of audibility	10	
	0		1	0·0002

Fig. 2. The volume of noise inside a standard Rover car is measured by means of a sound level meter on the front seat beside the driver

Fig. 3. Sound level testing in a Morris Minor two-door saloon

Fig. 1. A third network, giving an equal or flat response to sounds at all frequencies between 30 and 10,000 cycles per second at high noise levels, is incorporated. How this division covers the normal sounds met with in practice can be seen from the table.

With sounds of a fairly constant intensity, the meter deviates from the 'equal loudness' contour, to the human ear, by less than the divergence between different individuals of ordinary hearing. The response of the meter to sounds of an impulsive, or varying, nature is controlled by the inertia of the indicator used. This is a moving-coil instrument of a rapid-indicating type, the ballistic properties of which approximate to the equivalent characteristics of the human ear.

On a car there are, in general, two sources of noise, namely the engine, and the structure which, of course, includes the body. Noises from the engine are related to the speed of the machine under test, while the structure noises are not. Clearly engine noise will usually have a bigger energy content since there is a large and continuous source of energy behind it. Nevertheless, other noises can be more objectionable to the motorist, as they may well have a higher pitch and so sound more unnatural. The full-throated roar of a sports engine may be music in its owner's ears, whereas a soft squeak in the bodywork may annoy him beyond measure. Annoyance arising from body noise is probably accentuated because of uncertainty regarding its origin. It is most important, therefore, that these noises are eliminated.

The preference of the human ear for some sounds rather than others is most marked, and it is the fundamental aim of acoustical analysis to eliminate the unpleasant component of the noise.

Fig. 4. The bodywork of a Land-Rover is tested for resonance and vibration. The probe is used in conjunction with a sound level meter (nearer the operator) and an analyser

Fig. 5. Balancing the fan of a heater blower. Unbalance results in vibration, which is picked up by the vibration probe and indicated by the sound level meter

Fig. 6. Vibration meter mounted in the rear seat of a Consul for comparative tests on the efficiency of different engine mountings

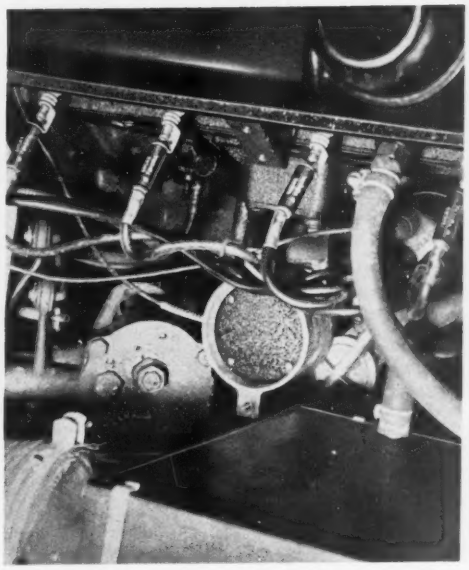


Fig. 7. One of the two vibration probes, see Fig. 6, in contact with the engine of the Consul at the point of suspension

This may be attempted in one of three ways. The first, of course, is to reduce the volume. Where this is impracticable, another possibility is the displacement of the frequency of the sound up or down the scale into those regions where the human ear is less sensitive. If the frequency of the noise can be moved above about 17,000 cycles per second, it becomes almost inaudible to the human ear. That it is still a sound, however, is established by the fact that a dog responds readily to a "silent" whistle which is pitched at a supersonic frequency too high for the human ear to perceive.

Noise is, however, not only unpleasant, but it may also indicate a degree of vibration that is loosening the components. That is why a quiet car usually wears better than one which has, from the commencement of its life, a metallic rattle. It also explains why an old car seems to develop squeaks and noises in many places almost simultaneously. Once looseness and vibration have started, they tend to spread fairly rapidly, the effect usually being cumulative.

This interdependence of sound and vibration means that vibration analysis can be very useful in acoustical work. Although vibration measurements with a probe type of pick-up cannot be made to include air-borne noises, such measurements are selective by virtue of the fact that the probe can be put into contact with the part whose audible output is required to be checked. The probe ignores background noise, and the meter records only the performance of the component under test. Measurements of sound conduction in structures, such as walls and doors, as well as in exhaust manifolds and other components, can be made very effectively in this way.

Fortunately, the commercial sound level meter, shown in use in Figs. 2 and 3, can readily be converted into a vibration meter by substituting for the microphone an inertia-operated piezo-electric vibration pick-up, feeding a signal to the meter by way of a vibration adaptor unit. This adaptation is seen in Figs. 4 and 5. The pick-up generates a voltage proportional to the acceleration of the vibrating body, and successive integrating networks in the adaptor unit produce voltages proportional to the velocity and displacement of the point under test. A three-way switch is employed to select the desired response from the three quantities of acceleration, velocity and displacement.

The lowest frequency level that can be handled when using the adapted sound-level meter is 25 cycles per second. A wider range from 2 to 1,000 cycles per second can be obtained from a special-purpose vibration meter. This meter is recommended not only for measurements at the lowest frequencies, but also where vibration measurement is the primary requirement.

In the automobile industry there are many applications, outside the acoustic field for vibration measurement. These include the testing of bearings of all types as well as the dynamic balancing of rotating parts. The smoothness of an engine can perhaps be better judged by this means than by any other. In this connection, Fig. 6 shows a typical set-up used, by Firestone Tyre and Rubber Co. Ltd., when carrying out research on engine mountings for the Ford Motor Co. Ltd. who themselves use similar equipment in their experimental department.

The vibration meter, in a container, is laid on the rear seat. This container is an open-topped wooden box, suitably lined with sponge rubber pads to reduce transmitted vibration to an absolute minimum. A writing table is mounted on the back of the front seat so that the readings may be recorded with convenience. A switch box enables either of two vibration probes, in contact with the engine at the point of suspension as at Fig. 7, to be connected to the meter. To exclude interference from the high tension circuit of the car, the probes are placed in copper containers. Successive recordings are made of both the horizontal and the vertical movements of the engine.

In order that truly comparative tests of the efficiency of different mountings may be carried out, all external conditions are maintained

constant. Pool petrol is used throughout with a standard carburettor setting, and the tyre pressure is checked frequently, and adjusted to the specified figure. A desirable feature is the provision of a specially calibrated revolution indicator, in this case it was mounted on the steering column, to give an accurate figure for the speed of the test car. The car in the illustration is a Consul off the normal production line. By this experimental work, it is hoped to develop even better rubber engine mountings, which will reduce vibration in the chassis to an absolute minimum and thereby increase passenger comfort.

Sound level and vibration measurements are used both for development of prototype cars and also for routine production testing. Actually, these two functions are complementary. A highest permissible noise level can be established from work on the prototype, and production models may then be checked to ensure that they do not exceed this figure.

Although they are complementary, prototype and production model tests have entirely different objects. The aim of this work on the prototype is, generally speaking, to reduce the noise level by discovering what features are producing the noise. On the other hand, production model tests are to check that the noise levels, found to be reasonable from tests on the prototype, are not exceeded.

It is most important that a sound level meter for car work should be compact and portable, so that it may be used on the road. Data, which would be impossible to get from test-bed work, can be obtained from road testing. A typical arrangement of a sound level meter used by the Rover Co. Ltd. is seen in Fig. 2. The meter and accessories, such as an analyser, are placed in any convenient position. No special mounting is necessary, though, as has already been stated, it is sometimes resorted to when testing for vibration. When measuring sound level, the microphone is placed at the point at which the noise level is to be measured. It is, of course, possible to mount the microphone in a position remote from the sound level meter; to take readings, for instance, near the engine or the exhaust manifold.

A useful accessory for use with the sound level meter is the analyser. The sound level meter itself will give a reading proportional to the total effect of the noise on the human ear, but it is sometimes required to find out what fraction of this total noise occurs at different frequencies. This is not only because, as already explained, noise at certain frequencies is more objectionable than at others, but also because it is the easiest way to track down noise to its source.

The Rover Co. Ltd. not only use sound and vibration meters in cars on road tests, but the noise level of the engine is also measured after an arduous test run, to check that it does not become noisy in service. Bodywork on the Land-Rover, by extensive testing using a vibration probe, Fig. 4, has been developed so as not to produce objectionable noises, whilst Fig. 5 shows a heater blower, used by this firm, being carefully checked for balance, with similar equipment. The Nuffield Organization also maintains the high reputation of its products by sound level testing. Fig. 3 shows a Morris Minor two-door Saloon with a sound level meter in use.

Another instrument that is very useful on actual trial runs is a "high speed level recorder". In a noise test being carried out on a car accelerating from a standing start to, say, 50 m.p.h. in a short space of time, the reading on the sound level meter will vary so quickly that it may be impossible for an observer to obtain an accurate record of the variation in sound level. Such a recorder will do this work automatically, and will produce a chart showing the variation over the whole of the period of the test. Not only is variation shown, but the chart can be readily calibrated directly in decibels in the same way as the sound level meter itself.

Acknowledgement is made to the Instrument Division of Dawe Instruments Ltd., of 83, Piccadilly, W.1, for permission to reproduce the illustrations of their equipment for sound level and vibration measurement shown throughout this article, and to the Technical Division of Firestone Tyre and Rubber Co. Ltd., of Brentford; to the Ford Motor Co. Ltd., Dagenham; Morris Motors Ltd., Cowley, and to the Rover Co. Ltd., of Solihull, for some of the information.

CHILLED CAST IRON TAPPETS

AN article in the *Eaton Forum*, April 1952, by P. Moss, describes the development of the Company's chilled cast iron tappet manufacturing methods from 1923 to date. Among the first refinements was the use of all-cast iron tappets instead of the cast iron head and steel body construction. Cupola cast iron was poured into sand moulds set on cast iron chill plates. As these produced an unsatisfactory finish of the chilled face, a solution was found in cutting the plates with a grillwork of shallow grooves. Chilled faces thus produced had a mottled structure and a Rockwell C hardness of 45-48.

The next improvement was twofold: (1) duplexing the iron from the cupola to an electric furnace where alloying elements such as Ni and Mo were added, and the iron superheated, and (2) the use of copper plates to obtain a better chill. The mottled areas were thus eliminated, the tensile strength raised to 40,000 lb/in^2, and Rockwell C hardness to 52-56.

Owing to post-war shortage of good raw materials, a new technique was developed which dispenses with the use of pig iron. The furnace charge consisting of steel scrap, cast iron returns, graphite and alloying additions such as ferro-silicon and ferro-molybdenum, is cold melted in a small rocking type electric furnace. The hard, dense chill structure was originally handicapped by a low strength due to a high Si content, but this disadvantage was later eliminated by altering the analysis of the iron and improving the melting practice. At the same time copper chill plates were replaced by carbon ones. The former, although excellent as heat conductors, warped in constant use and tended to produce an irregular chill. In comparison with a cupola chilled iron, the material shows a finer structure and greater freedom from graphite areas.

The examination of tappet sections reveals a desirable uniform and clear-cut demarcation line of the chilled zone. Consequently, on the one hand there is no danger of excessive local thinness of the layer after grinding, while, on the other hand, the central hole may be drilled to a maximum depth without risking the destruction of the drill. A high-quality, well-conditioned moulding sand with correct moisture content uniformly distributed is essential in the casting process. (*M.I.R.A. Abstract No.* 5881.)

Coachbuilding Adhesives

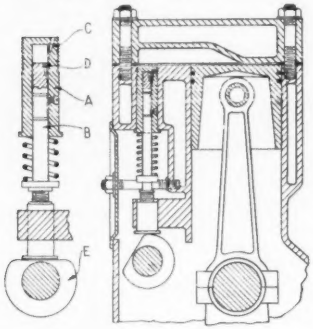
WRITTEN in non-technical language, a new booklet issued by Dunlop Special Products, Ltd., Fort Dunlop, Erdington, Birmingham, 24, describes a wide variety of adhesives available for modern coachbuilding applications. Bonding characteristics and methods of handling are outlined for solutions of natural rubber, reclaimed rubber, latex dispersions and special self-vulcanizing solutions. The booklet is well illustrated by photographs taken in the trim shops of a number of motor manufacturers. (2017)

CURRENT PATENTS

A Comprehensive Review of Recent Automobile Specifications

Compression ignition of petrol engines

WITH the object of eliminating electrical equipment this engine is ignited by the combustion of a small pilot charge which in turn fires the main charge in the working cylinder. The specification gives no indication of means to vary the timing of the ignition point. For a side-valve engine, the device comprises a compression cylinder *A* in which reciprocates a plunger *B* actuated through an adjustable tappet by the engine camshaft and



No. 669456

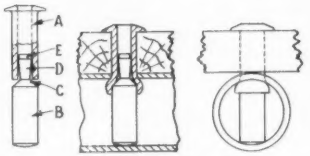
retracted by a spring. Cylinder and piston are ported at *C* and *D* respectively; the arrangement being that port *C* is uncovered by the plunger at the bottom of its stroke and the two ports are in communication when the plunger is at the top of its stroke. Port *C* is open to the main combustion space.

In operation, as the engine piston rises on the compression stroke, cam *E* allows the plunger to fall under the constraint of its spring until it opens port *C* and mixture flows into cylinder *A*. While the engine piston continues to rise, the plunger is urged by cam *E* to close port *C* and compress the charge in cylinder *A*. Timing is so arranged that at, or slightly before, the instant for ignition, ports *C* and *D* establish communication and the burning gases in cylinder *A* escape to ignite the main charge in the working cylinder.

Similar equipment operated by a push rod and rocker is proposed for overhead valve engines. *Patent No.* 669456. *F. Lawton.*

Riveted connection to tubular element

A RIVETED joint between two or more components of which one is a tube is effected by a hollow rivet and an anvil member which can be inserted and clinched from one side of the assembly. Hollow rivet *A* has a plain tubular shank while the anvil member *B* comprises a cylindrical body with a rounded free end,



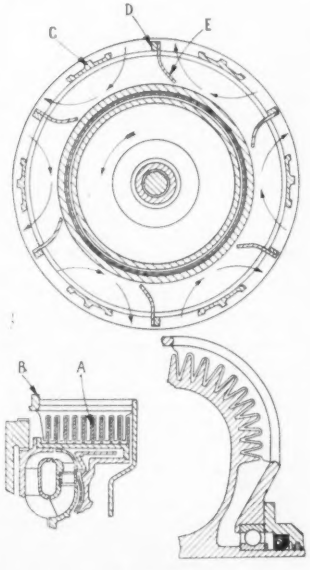
No. 669880

a conical transition part *C*, a conical stem *D* increasing in diameter from *C*, and a conical centering or guide part *E*. The combined length of the anvil body and part *C* is approximately equal to the inner diameter of the tube.

To assemble, the stem of the anvil is inserted in the shank of the rivet and together they are passed through holes in the parts to be joined; in the example, a board to a tube. When the rivet is pressed home, the shank deforms and is expanded over the conical part *C* of the anvil to form a head and bring the shank of the rivet under tension. *Patent No.* 669880. *De Cirkel N.V. and M. A. van der Rijst (Holland).*

Cooling hydrodynamic units

TORQUE converters and hydraulic couplings necessarily operate at less than 100 per cent efficiency and the power loss is apparent as heat that must be dissipated. Under certain operating conditions the quantity of heat to be with-

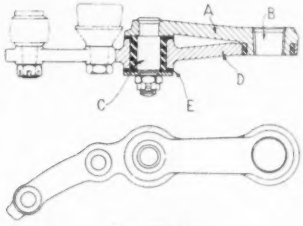


No. 667565

drawn is relatively large. Cooling effected by continuously passing a portion of the operating fluid through an external heat exchanger and then returning it to the hydraulic circuit increases cost, weight, and installation problems.

Air cooling by means of finned surfaces on a rotating element is likely to be inadequate as skin friction accelerates the immediately surrounding air and thus lowers the relative velocity and the circulating quantity. It is proposed, therefore, to provide a controlled air flow at a high relative velocity, to ensure a high rate of heat transfer, while at the same time requiring only a low absolute velocity, to minimize the expenditure of energy in accelerating and decelerating the ambient air.

The rotating pump element of the unit



No. 669245

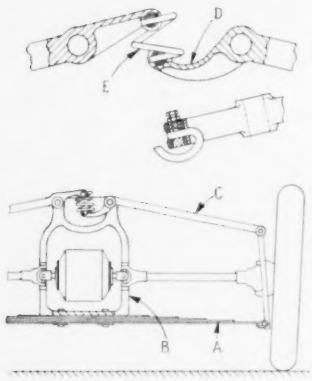
is furnished with a series of circumferential fins *A* and surrounded by a stationary cage-like structure *B* having a plurality of axially disposed webs *C* alternated with ribs *D*. To each of these ribs is attached a vane member *E* with inwardly projecting fingers suitably contoured to substantially fill the grooves between fins *A*. In association, webs, ribs and vanes provide two series of openings for the entry and discharge of air to and from the grooves between the fins. An alternative arrangement, particularly suitable for fluid couplings, is illustrated.

In a quoted example a torque converter transmitting about 50 h.p. and having a peripheral speed of 40 m/sec was adequately cooled in this manner and the power consumption was only approximately 0·25 h.p. *Patent No.* 667565. *AB Ljungstroms Angturbin (Sweden).*

Resilient drop arm

TO insulate the steering gear and steering wheel from road shocks transmitted through the linkage, this divided drop arm is coupled through two interposed rubber sleeves. Lever *A* is secured to the steering shaft by means of a tapped hole *B*, it is stated, and at its other end carries a shouldered stud *C*.

On a rubber sleeve on this stud is mounted the hub of lever *D*, flanked by metal washers *E* and retained by a nut and a thick washer. The end of one arm of lever *D* encircles, with an interposed rubber sleeve. the hub of lever *A*, while to the other arm are connected the steering links. *Patent No.* 669245. *Fiat Societa per Azioni (Italy).*



No. 669545

Anti-rolling device

IN a suspension system having independently suspended and independently sprung wheels, transverse wheel arms, one on each side of the vehicle, are extended to form levers. The free ends of the levers are spaced one above the other and coupled by a helical spring to prevent or reduce rolling when cornering at speed. The example illustrated shows a suspension parallelogram formed by half of a transverse spring *A*, secured at its centre to the vehicle framing *B*, and a pivoted arm *C*. Integral with arms *C* are levers *D* arranged at a suitable spacing and connected by a short spring *E* having only one or two coils. The ends of the spring, arranged parallel to each other, are threaded through journal blocks bearing in the ends of the levers.

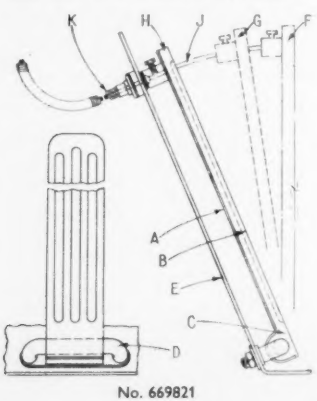
When both wheels have the same vertical movement the spring *E* is unaffected but with unequal wheel movement the spring acts, by tension or by compression, to oppose the tendency to roll. *Patent No* 669545. *J. Gregoire (France).*

Accelerator pedal

THIS control pedal is formed of a sheet metal back plate *A* terminating in a half sleeve. To the back plate is bonded a rubber tread *B* with an integral sleeve *C*. Also bonded to sleeve *C* is the central limb of a stout U-bolt *D*, by which the pedal is pivotally anchored to the floor or toe board *E*.

In a free condition the pedal would occupy the position indicated at *F* but in the normal closed position of the carburettor throttle it will be at *G*. Accordingly, the rubber sleeve *C* is always torsionally stressed. The torsion increases as the pedal is moved to the fully open position *H*, determined by a stop.

Attached to the end of the pedal is a push-pull cable *J*, preferably guided in a coiled sheath *K* to ensure a flexible lay. *Patent No.* 669821. *Teleflex Products Ltd., J. S. Bentley, George Spencer Moulton and Co. Ltd. and A. E. Moulton.*

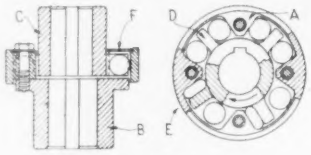


No. 669821

Torsionally resilient shaft coupling

THE view is advanced that many types of resilient couplings are insufficiently elastic, or necessitate accurate alignment of shafts to ensure satisfactory operation. Rubber balls have been used as the elastic elements but are commonly restricted in action by concave walls or abutments.

The illustration shows an eight-ball coupling of the proposed type, in which the abutments *A* on sleeve member *B* are attached as a separate unit, but in a four-ball coupling, also described in the specification, the abutments are formed integrally. Sleeve *C* is furnished with radial vanes *D* and between each vane and abutment is located a rubber ball. It will be noted that the face of the abutment is so arranged that when the coupling is under full load, as indicated in the scrap section at *E*, it is substantially parallel to the face of the vane.



No. 669724

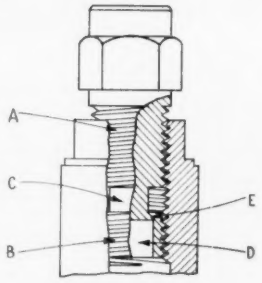
A sheet metal cover plate *F*, attached by the screws securing the abutment ring, retains and encloses the rubber balls. *Patent No.* 669724. *J. Holoye (Belgium).*

Self-locking tappet screw

INTENDED primarily, but not exclusively, for tappet adjustment, this self-locking screw has a shank comprising two mismatched screw portions spaced by a plain stem of reduced diameter and a diaphragm extending in a plane perpendicular to the axis.

Referring to the drawing, the hexagon head is formed with a bearing surface to engage a valve stem in the usual manner, and the shank has two screwed sections *A* and *B* of like hand and equal pitch. The threads may be produced economically by rolling, with the respective dies axially offset to one another. Joining the two screwed sections is a coaxial stem *C*, and section *B* is machined with an axial bore *D* terminating a short distance from the end of the stem to leave a connecting diaphragm *E*. This diaphragm provides a predetermined resilience between the screwed sections *A* and *B*, sufficient axial deflection for engagement, low torsional deflection and a satisfactory elastic return.

While the mismatching of the screwed sections must be within certain limits, it is not a critical factor as the torque characteristics of the screw are determined by the thickness of the diaphragm and the diameter of the reduced stem. Depending upon the application, the mismatching of the screwed sections may be either negative or positive. *Patent No.* 669698. *Eaton Manufacturing Company (U.S.A.).*



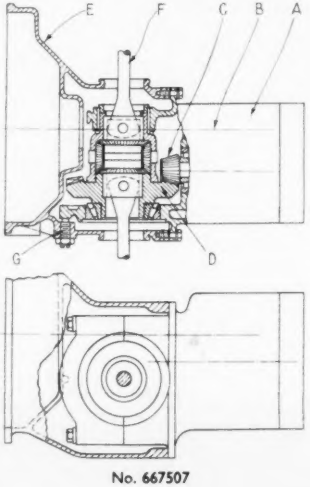
No. 669698

Front-wheel drive layout

THIS arrangement of a transverse drive located between engine and gear units, permits the final bevel gearing to be adjusted precisely before assembly and also gives opportunity to save weight in the casings. The change-speed gearbox *A* at the rear is driven from the clutch shaft on axis *B* and returns the drive through bevel pinion *C*.

Crown wheel *D* and the differential gearing are supported in bearing housings integral with the cast iron gearbox and completed by caps secured by setbolts. They can thus be assembled and adjusted on the gear unit in a freely accessible and observable disposition. The light alloy clutch housing *E* is extended to enclose the transverse drive and is flanged at the rear to receive the gearbox and apertured on each side to admit the axle shafts *F*.

Although the axle gearing bearings are relatively rigid, additional support is provided to counteract the axial thrust of the crown wheel by a screw *G* which is adjusted in housing *E* to engage the adjacent bearing cap. *Patent No.* 667507. *Daimler-Benz Aktiengesellschaft (Germany).*



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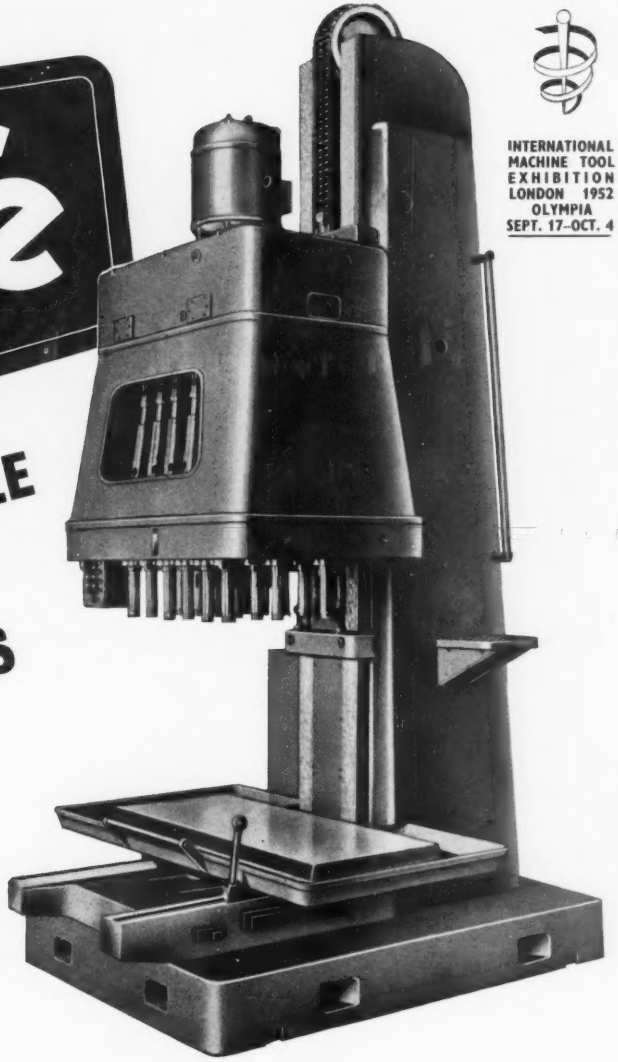
INTERNATIONAL MACHINE TOOL EXHIBITION LONDON 1952 OLYMPIA SEPT. 17–OCT. 4

MULTI-SPINDLE DRILLING MACHINES



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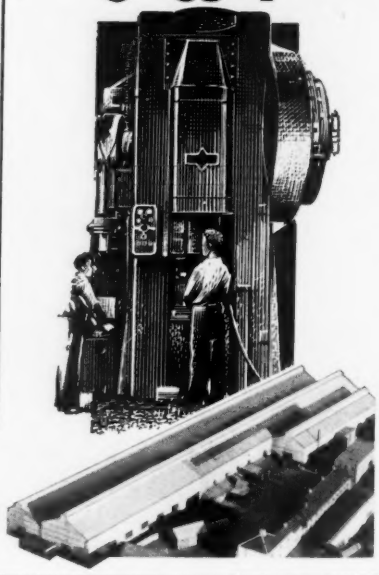

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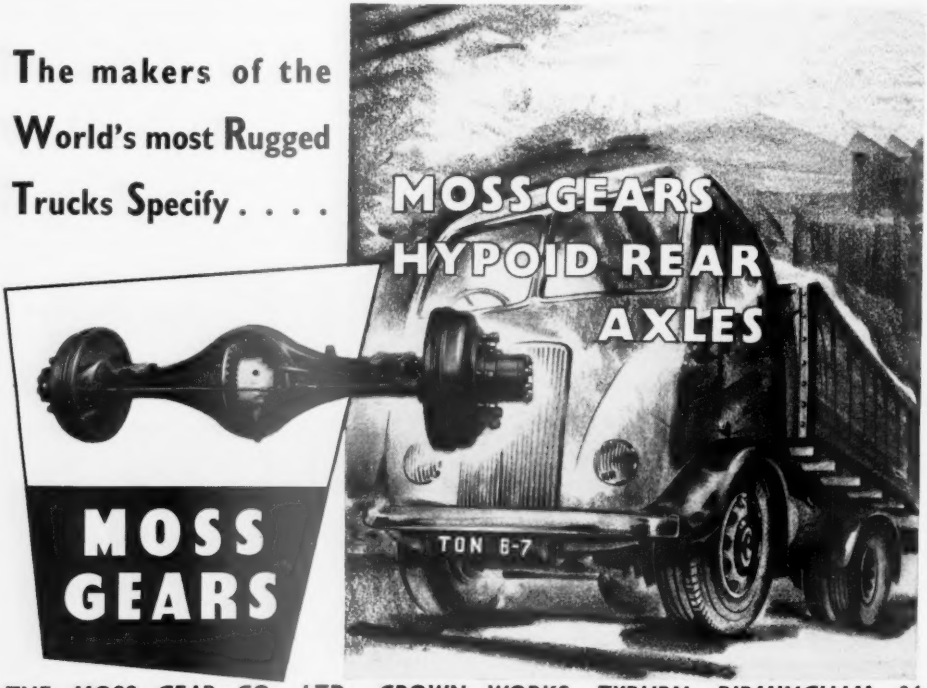

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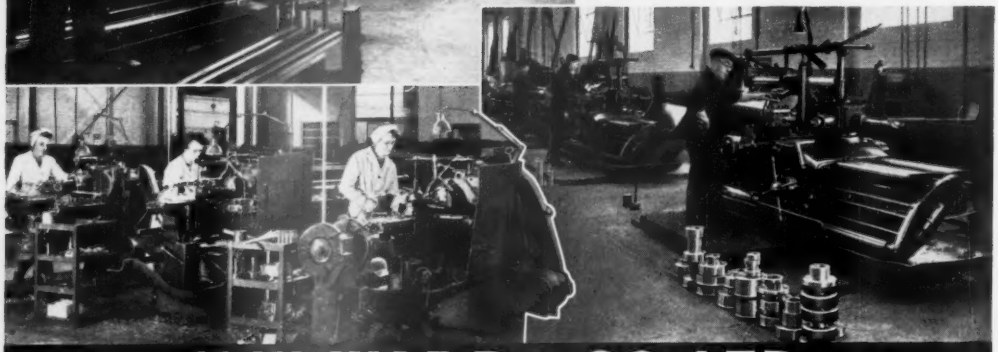

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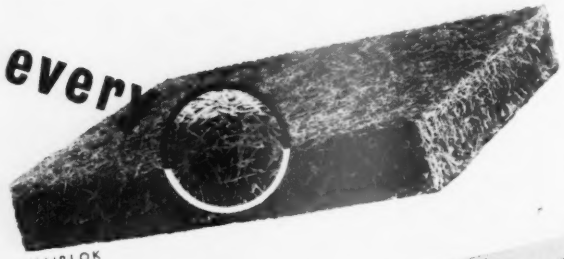

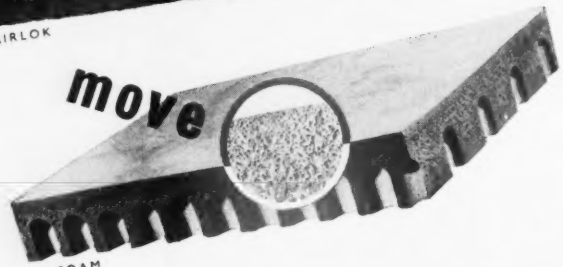

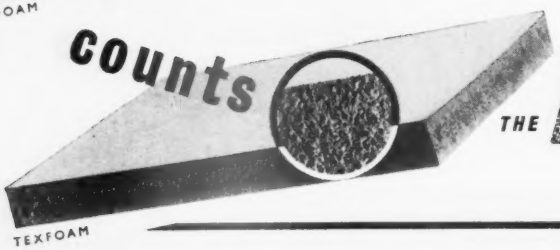

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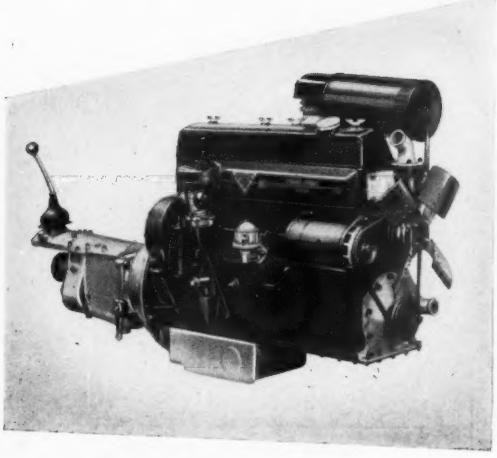

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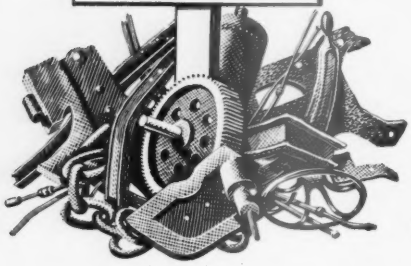
WHAT EVERY MANAGEMENT SHOULD KNOW ABOUT

scrap



What is scrap?
Obsolete plant and machines, old spares, redundant buildings—anything made from iron and steel for which you have no foreseeable use.

What good is it to me?
No good at all in its present state. But send it back to the steelworks and it can go to make the new steel that you and all steel-users need. Remember—a ton of scrap may make a ton of new steel.

Can't they make steel without scrap?
Yes—but not enough steel. Last year, 58% of our steel was made from scrap. The more scrap we use the more coal, coke and iron ore we save for other important purposes.

What should I do with my scrap?
See your local scrap merchant. He will be glad to help with dismantling and collection. If you have any trouble disposing of iron and steel scrap, write to the British Iron & Steel Federation for the address of your Joint District Scrap Committee.

What shall I get for it?
You will be paid a fair price for your scrap. MORE IMPORTANT—you will be doing yourself and all steel-using firms a great service by helping to increase steel production.

SPEED THE SCRAP

SPEED THE STEEL

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CLAYTON

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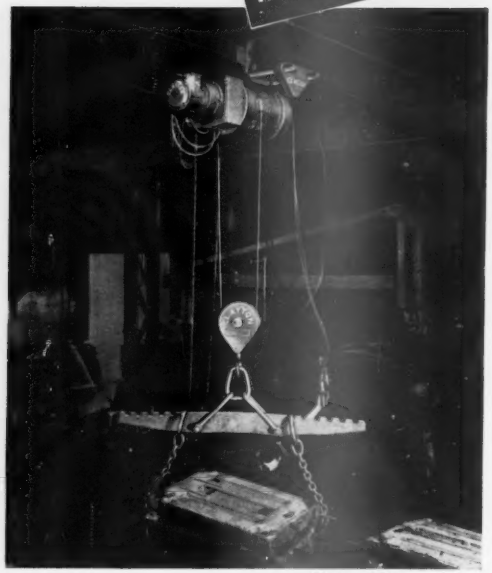

Photograph by courtesy of Newcast Foundries Ltd., Newcastle, Staffs.

Besides hoisting, lowering and travelling at ordinary speeds, CLAYTON foundry hoists with MICRO-SPEED unit, provide for reliable and precise dead-slow movement so essential in the foundry (and for other precision work). The normal A.C. supply is used.

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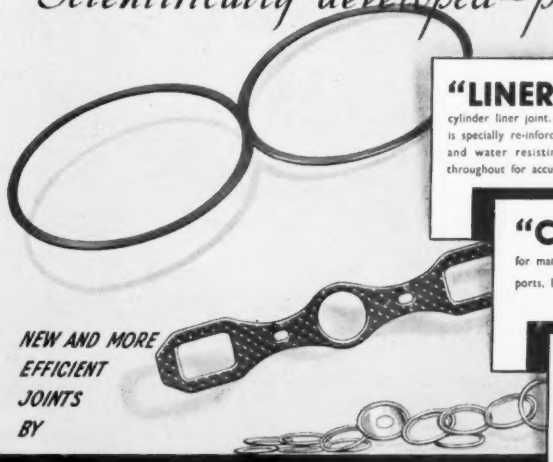

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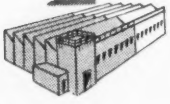

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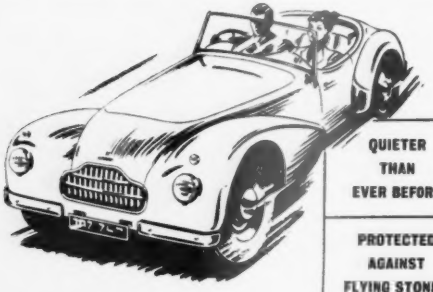
Rubberised PROTECTIVE COATING

IS SO IMPORTANT TO THE LIFE OF ALL CARS THAT IT SHOULD BE APPLIED BEFORE CARS FIRST TAKE THE ROAD!



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PROTECTED AGAINST FLYING STONES

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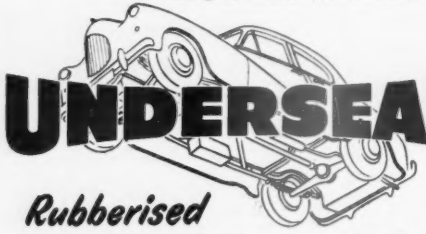

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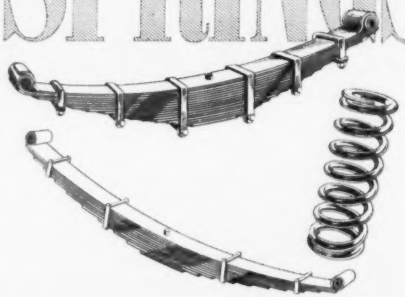
Rubberised PROTECTIVE COATING

adds years to the life of a car

SPRINGS

for the motor industry



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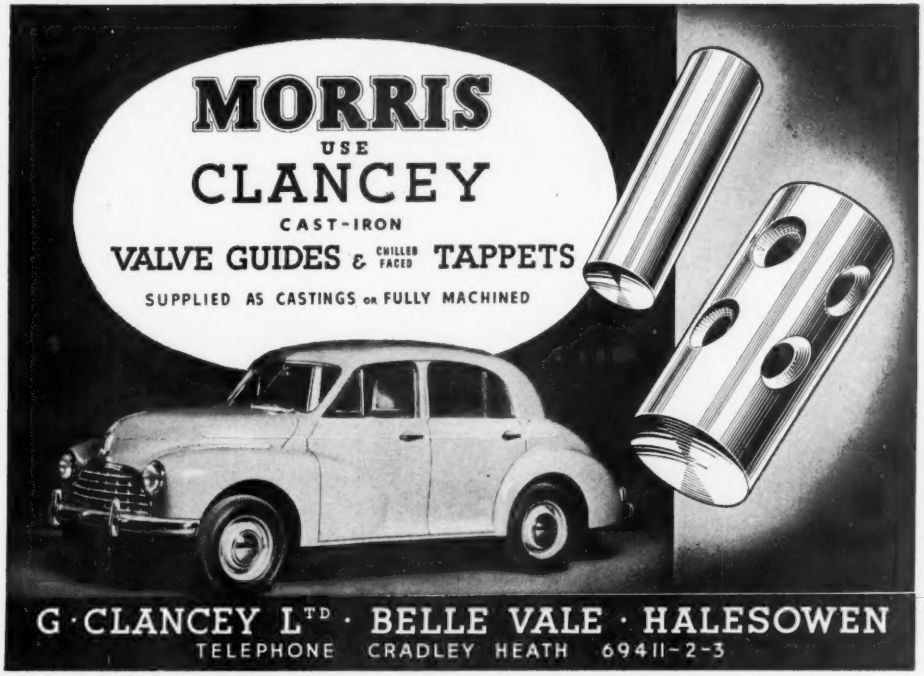
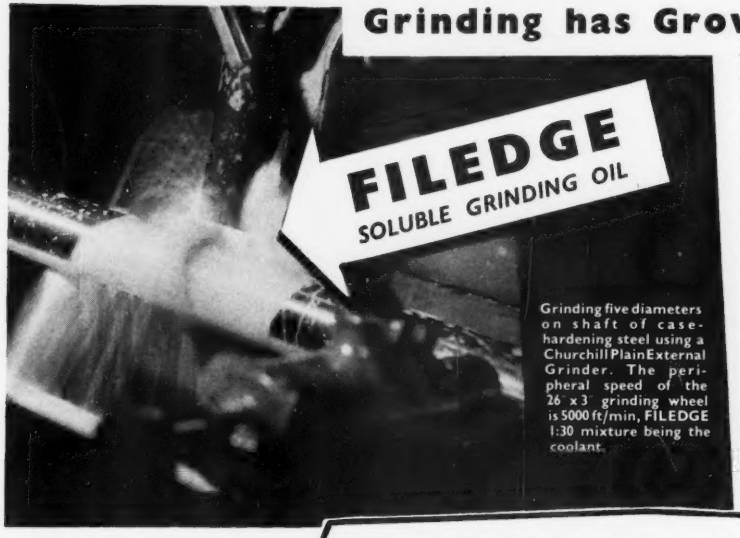


Grinding has Grown Up!



Grinding five diameters on shaft of case-hardening steel using a Churchill Plain External Grinder. The peripheral speed of the 26" x 3" grinding wheel is 5000 ft/min, FILEDGE 1:30 mixture being the coolant.

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Photograph shows I-C engine pistons treated with 'dag' colloidal graphite before running-in. (Reproduced by courtesy of The British Piston Ring Co. Ltd.)

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Many processes involve high temperatures that destroy conventional lubricants but **'dag'** colloidal graphite, brushed or sprayed, forms a graphite surface on hot moving parts which provides effective lubrication at temperatures up to 600°C in air, giving longer life to moving parts and reduced maintenance cost.

Illustration of graphited camshaft by courtesy of The Standard Motor Company Limited.

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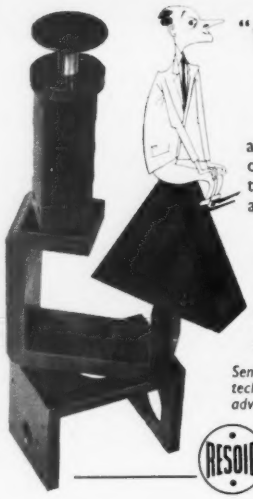

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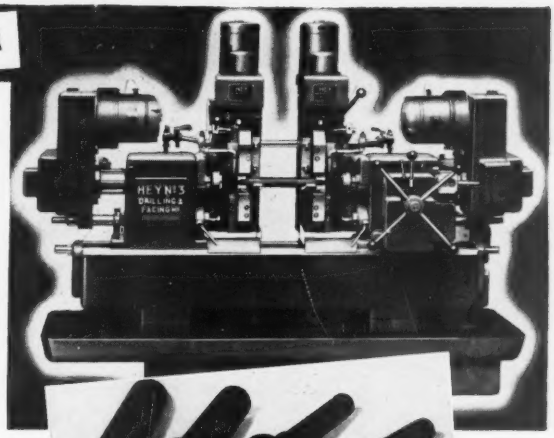

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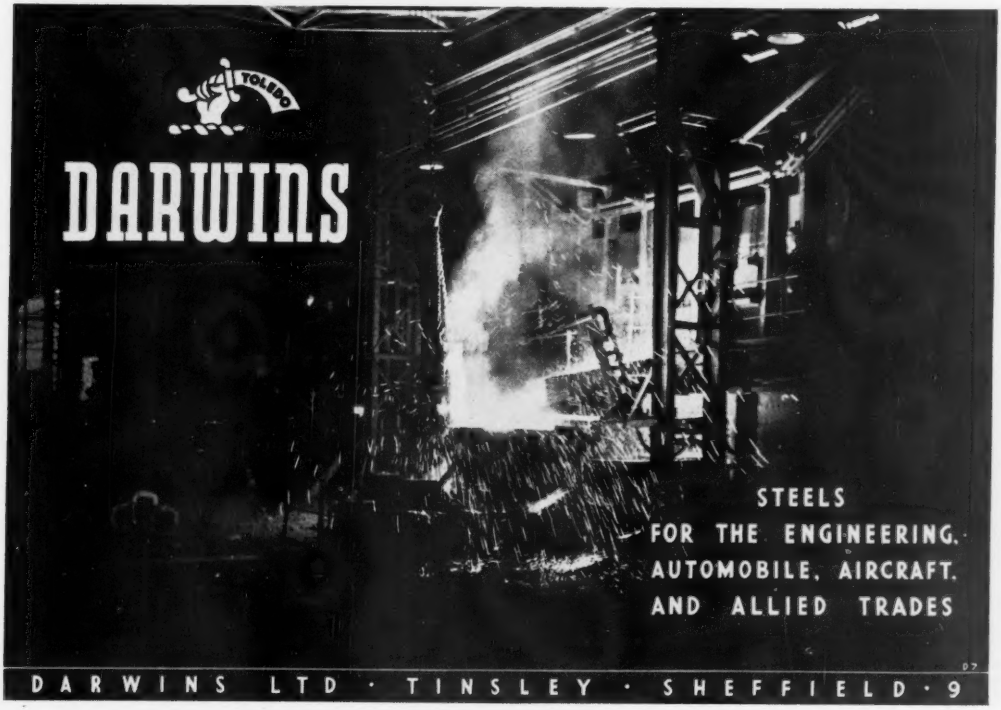

Air-cooled aero-engine cylinder showing the bore Monochrome processed with perfect oil-control provided by the HONEY-CHROME cellular surface. Photo by courtesy of the Ministry of Supply.

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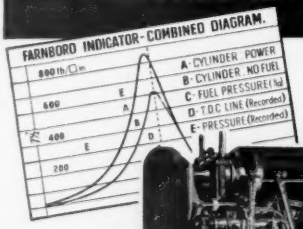

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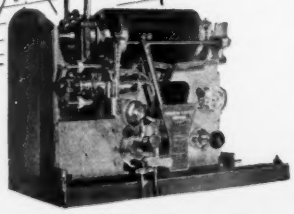

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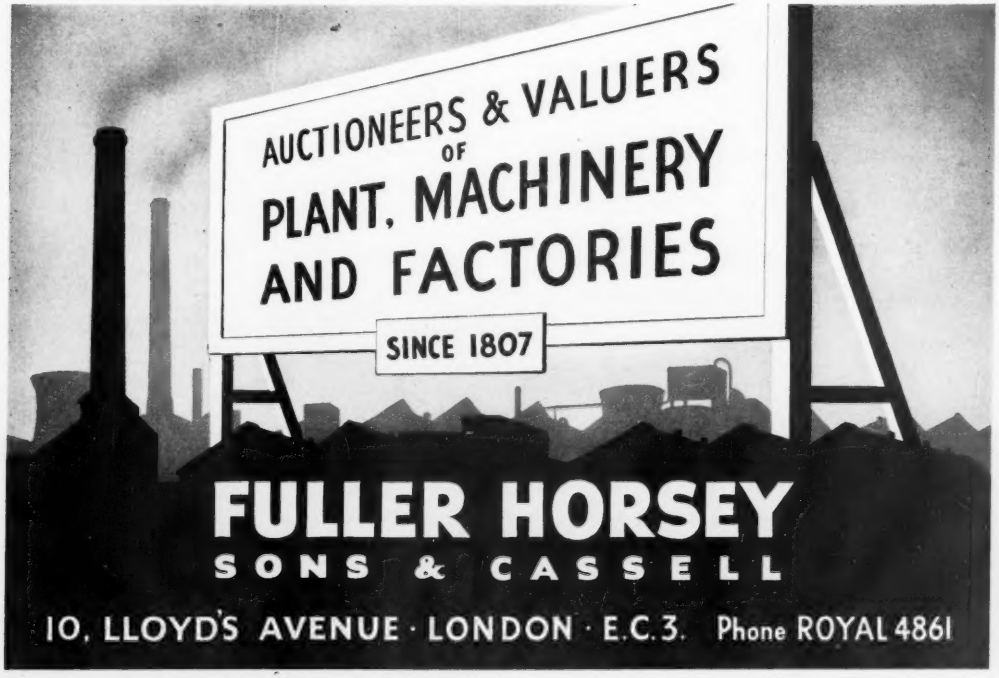

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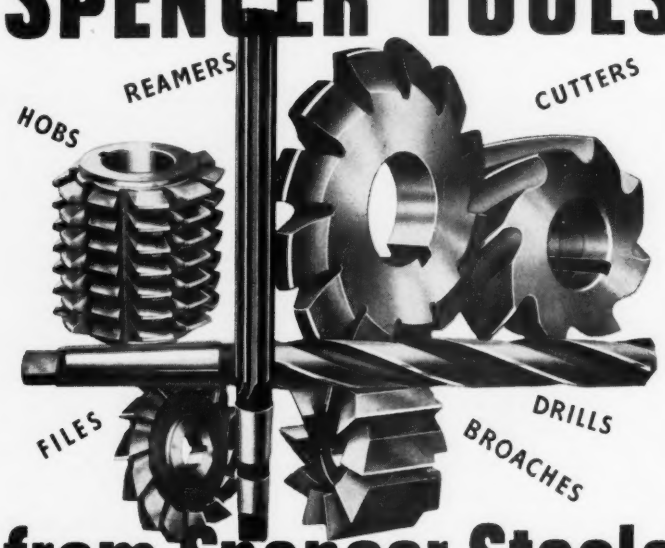

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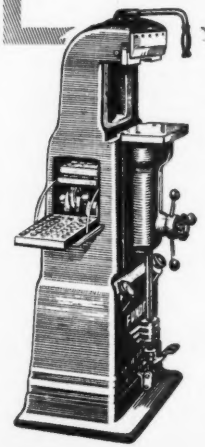

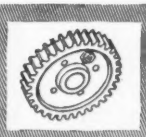
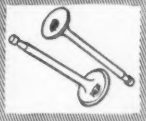

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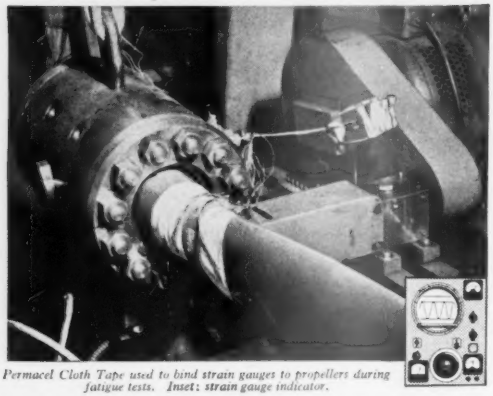

Permacel Cloth Tape used to bind strain gauges to propellers during fatigue tests. Inset: strain gauge indicator.

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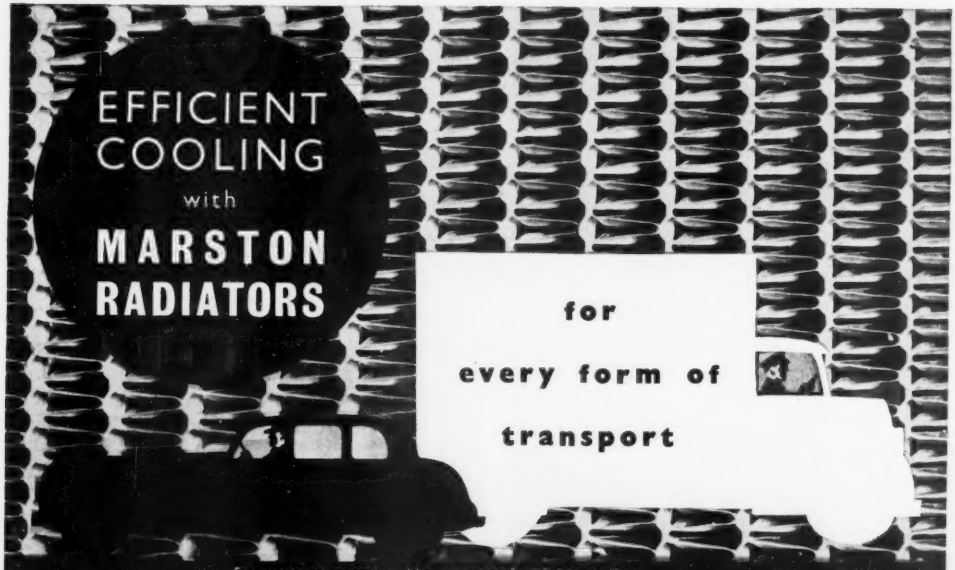

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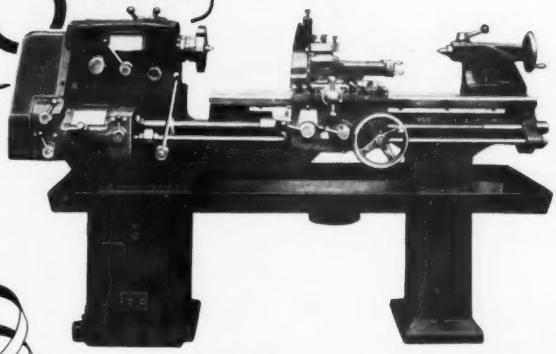
MAR. 94

wm 70 JUNIOR — 7" CENTRE LATHE



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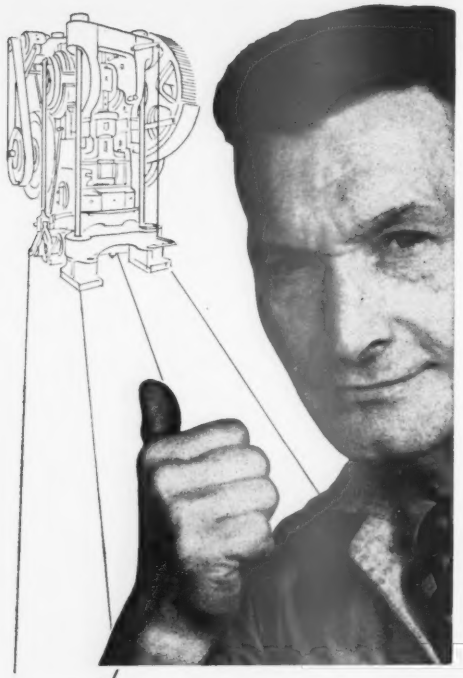

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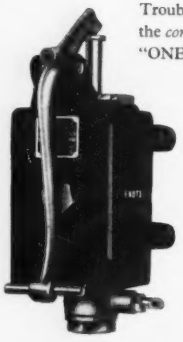

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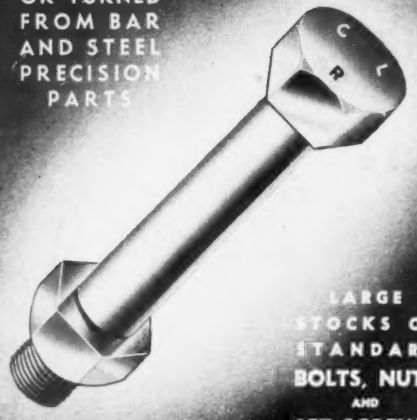

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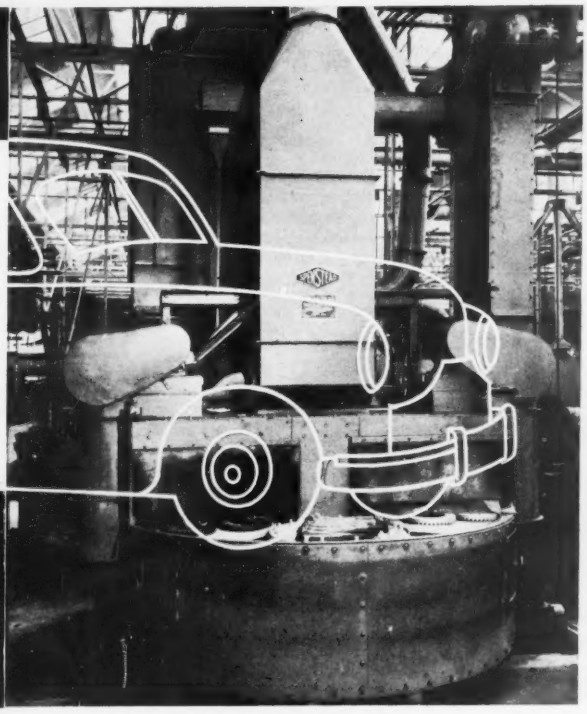

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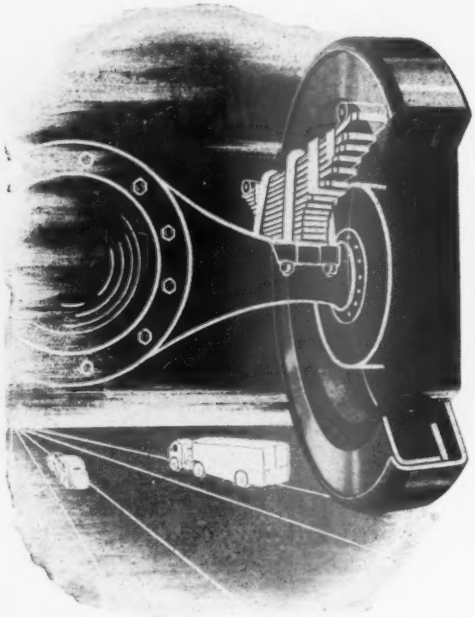

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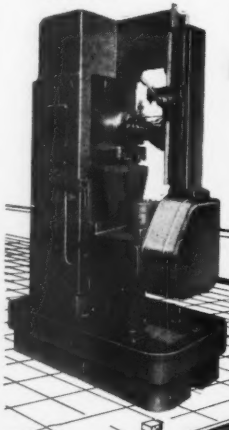

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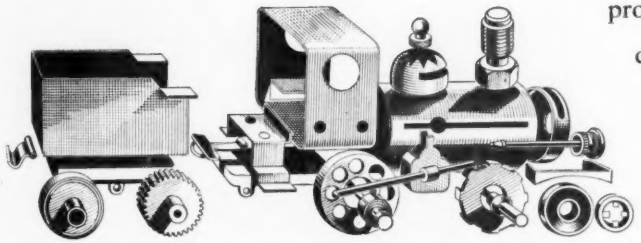

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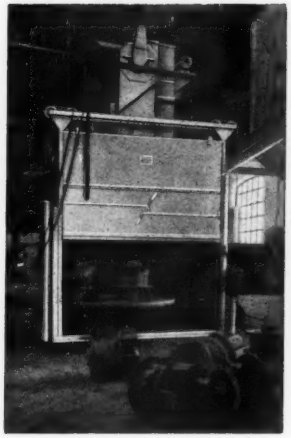

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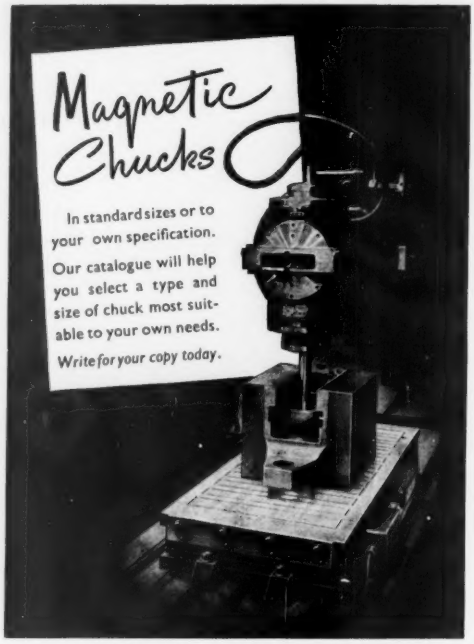
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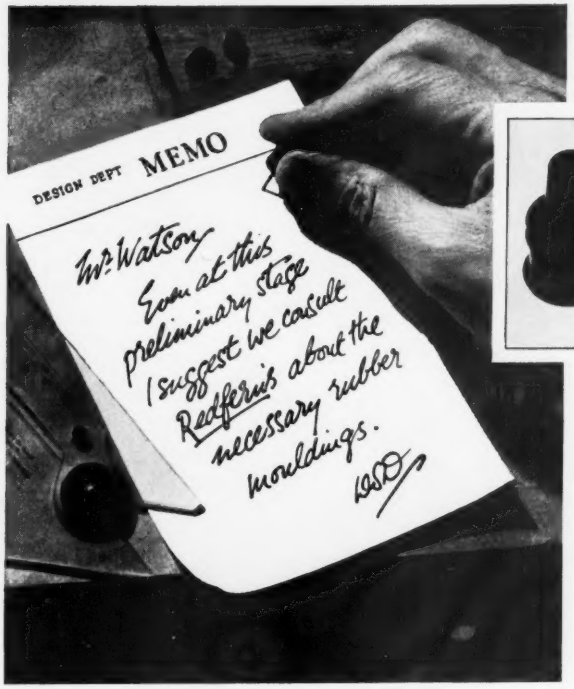
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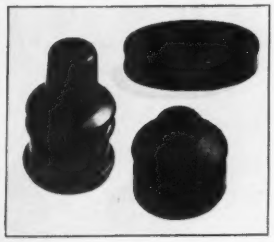

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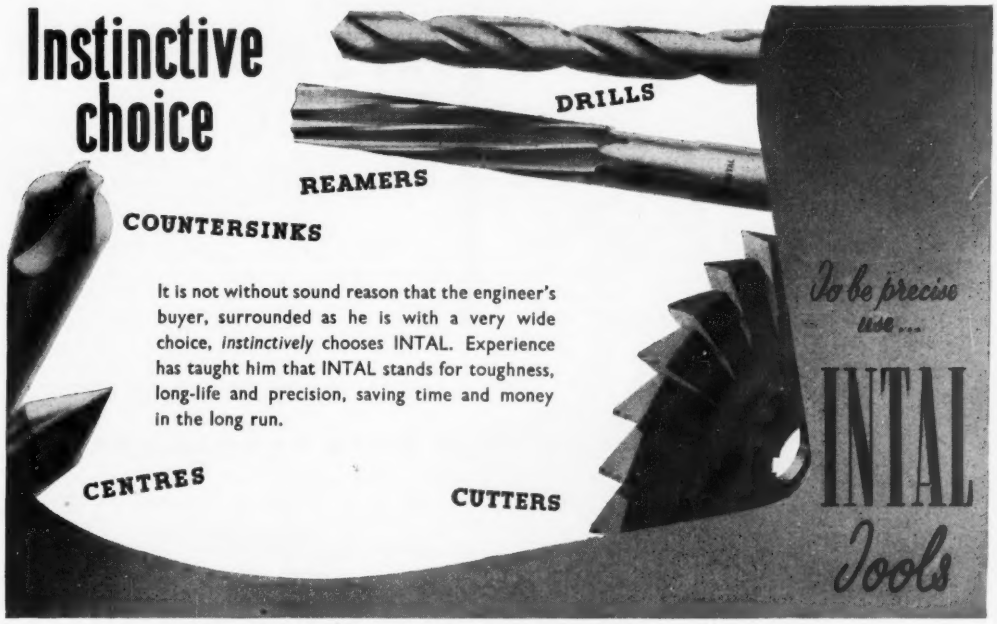

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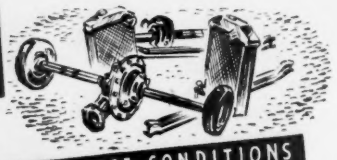

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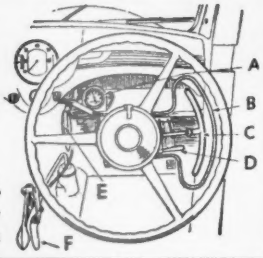

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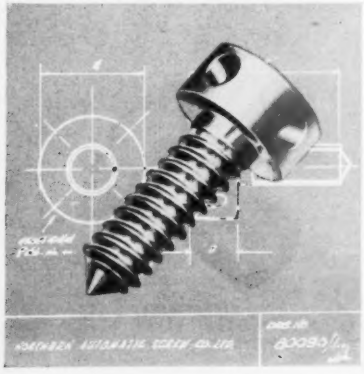

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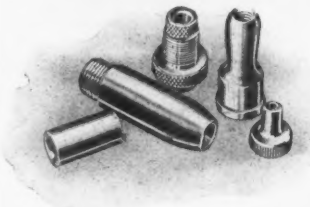

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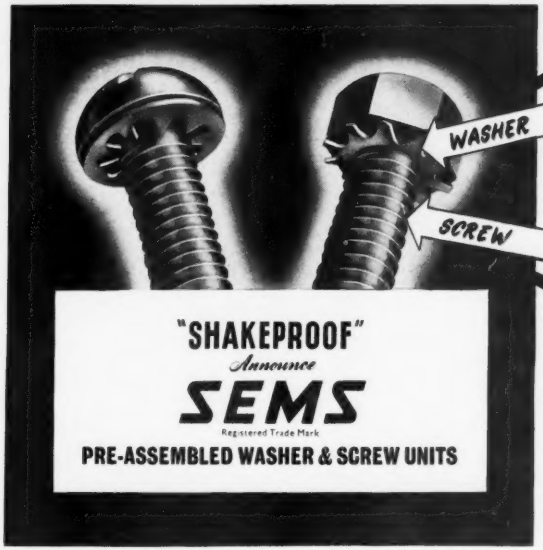

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INDEX TO MANUFACTURERS' ANNOUNCEMENTS

	PAGE
AC Sphinx Spark Plug Co.	24
Acheson Colloids, Ltd.	98
Acratork Engineering Co. Ltd.	106
Adamant Engineering Co., Ltd.	79
Aluminium Bronze Co., Ltd.	117
Aluminium Die Casting (Birmingham), Ltd.	106
Amal, Ltd.	92
Andrews, Harold, Grinding Co., Ltd.	78
Andre Rubber Co., Ltd.	57
Angus, George, & Co., Ltd.	45
Archdale, James & Co., Ltd.	34
Automotive Products Co., Ltd.	9, 10, 11, 12
Barber & Colman, Ltd.	119
Benton & Stone, Ltd.	106
Birlec, Ltd.	73
Birmingham Aluminium Casting (1903) Co., Ltd.	Cover iv
Booth, James & Co., Ltd.	49
Bound Brook Bearings (G.B.), Ltd.	82
Bridges, S. N. & Co., Ltd.	74
British Electrical Development Association	32
British Iron & Steel Federation	92
British Oxygen Co., Ltd., The	93
British Wedge Wire Co., Ltd.	116
Broom & Wade, Ltd.	80
Burton, Griffiths & Co., Ltd.	22
Cape Asbestos Co., Ltd., The	42, 43
Carr Fastener Co., Ltd.	112
Cary, Wm. E., Ltd.	96
Cashmore, John, Ltd.	108
Cassell Cyanide	38
C.A.V., Ltd.	40
Churchill, Charles, & Co., Ltd.	102, 110
Clancey, G., Ltd.	97
Clayton Crane & Hoist Co., Ltd., The	94
Clayton Dewandre Co., Ltd.	55
Cleveland Petroleum Co., Ltd.	54
Clifford, Charles & Son, Ltd.	107
Climax Molybdenum Co. of Europe, Ltd.	68
Cocker Bros., Ltd.	71
Cole, E. K., Ltd.	114
Collier & Collier, Ltd.	102
Cooper & Co. (Birmingham), Ltd.	114
Coopers Mechanical Joints, Ltd.	95
Cossor, A. C., Ltd.	Cover iii
Coventry Motor Fittings Co., Ltd.	27
Dale, John, Ltd.	75
Darwins, Ltd.	99
Desoutter Bros., Ltd.	17
Dobbie, McInnes, Ltd.	100
Dunford & Elliott (Sheffield), Ltd.	104
Dunlop Rubber Co., Ltd.	61
Eagle Pencil Co., Ltd., The	118
Eaton Axles, Ltd.	53
Edwards, F. J., Ltd.	113
Ein-Tout-Can Co., Ltd.	96
English Electric Co., Ltd.	62
English Steel Corporation, Ltd.	Cover ii
Feeny & Johnson, Ltd.	116
Feltham, Walter, H. & Sons, Ltd.	118
Ferodo, Ltd.	26
Firth Brown Tools, Ltd.	5
Fletcher Miller, Ltd.	97
Fuller, Horsey, Sons & Cassell	101
Funditor, Ltd.	103
Gas Council, The	13
General Electric Co., Ltd.	56, 117
Girling, Ltd.	51
Graham, H. G., & Son, Ltd.	108
Griffiths, Gilbart, Lloyd & Co., Ltd.	111
Groves, Samuel, & Co., Ltd.	112
G.W.B. Electric Furnaces, Ltd.	83
Habershon, J. J. & Sons, Ltd.	119
Hairlok Co., Ltd., The	90
Hardy, Spicer & Co., Ltd.	16
Harper, John & Co., Ltd.	91
Herbert, Alfred, Ltd.	19
Hey Engineering Co., Ltd.	99
Hoffmann Manufacturing Co., Ltd., The	15
Holroyd, John & Co., Ltd.	29
Humphreys, J. H. & Sons	112
I.C.I. (Marston Excelsior, Ltd.)	105
Imperial Smelting Corporation, Ltd.	77
Insley (London), Ltd.	118
International Twist Drill Co., Ltd., The	115
Iso-Speedic Co., Ltd., The	25
Jackson, H., Ltd.	21
Jenks Bros., Ltd.	104
Jessop, William, & Sons, Ltd.	52
Johansson, C. E., Ltd.	101
Johnson & Johnson (Great Britain), Ltd.	104
Kendall & Gent, Ltd.	88
Kirkstall Forge Eng., Ltd.	4, 92
Lewis Spring Co., Ltd.	94
Leys Malleable Castings, Ltd.	23
Lindley, C., & Co., Ltd.	106
Linread, Ltd.	30
Machine Tool Exhibition, The	31
Marbaix, Gaston E., Ltd.	33
Marshall, C. & C., Ltd.	100
M.C.L. and Repetition, Ltd.	118
Mek-Elek Engineering, Ltd.	118
Metalastik, Ltd.	69
Midland Mechanical Developments, Ltd.	67
Midland Motor Cylinder Co., Ltd., The	Cover i
Mills, William, Ltd.	120
Minnesota Mining & Manufacturing Co., Ltd.	96
Mond Nickel Co., Ltd., The	28
Monks & Crane, Ltd.	7
Monochrome, Ltd.	100
Morgan Crucible Co., Ltd., The	70
Moss Gear Co., Ltd.	86
Murad Developments, Ltd.	46
Nash & Thompson, Ltd.	90
Neill, James, & Co. (Sheffield), Ltd.	14
Newall Group Sales, Ltd.	58
Newall, A. P. & Co., Ltd.	87
Nitralloy, Ltd.	116
Northern Automatic Screw Co., Ltd.	116
Perry Barr Metal Co., Ltd.	95
Ransome & Marles Bearing Co., Ltd.	39
Ransomes, Sims & Jefferies, Ltd.	64
Redfern's Rubber Works, Ltd.	113
Renfrew Foundries, Ltd.	81
Renold & Coventry Chain Co., Ltd., The	6
Resinoid & Mica Products, Ltd.	98
Rockwell Machine Tool Co., Ltd.	85
S. & D. Rivet Co.	94
Sankey, Joseph & Sons, Ltd.	20
Seal-Less Strapping Ltd.	98
Sheffield Twist Drill & Steel Co., Ltd., The	37
Shell-Mex & B.P., Ltd.	18
Silentbloc, Ltd.	41
Simms Motor Units, Ltd.	48
Skefko Ball Bearing Co., Ltd. The	84
Slick Brands, Ltd.	93
Small & Parkes, Ltd.	66
Smethwick Drop Forgings, Ltd.	59
Solus-Schall, Ltd.	102
South Wales Forgemasters, Ltd.	86
Spencer & Halstead, Ltd.	109
Spencer, Walter & Co., Ltd.	103
Steel Company of Wales, Ltd., The	36
Sterling Metals, Ltd.	72
Tecalemit, Ltd.	35
Tempered Spring Co., Ltd., The	2
Terry, Herbert, & Sons, Ltd.	8
T.I. Aluminium, Ltd.	63
Tilghman's Patent Sand Blasting Co., Ltd.	111
Toledo Woodhead Springs, Ltd.	60
Tube Products, Ltd.	65
Udal, J. P., Ltd.	107
Vandervell Products, Ltd.	47
Vaughan, Edgar, & Co., Ltd.	89
Walker, M. W., & Staff, Ltd.	50
Ward, H. W. & Co., Ltd.	88
Ward, Thomas W., Ltd.	105
Watveare Overseas, Ltd.	109
Weston, Charles & Co., Ltd.	110
Whyte & Collins, Ltd.	110
Wickman, Ltd.	1, 3
Wiggin, Henry & Co., Ltd.	44
Woodhead, Jonas, & Sons, Ltd.	76
Yarwood, Ingram & Co., Ltd.	115

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